September 1932

Heating · Piping and Air Conditioning

Vol. IV No. 9

• In order to insure clean air in the Detroit Edison Company's three-story research building, it was necessary to provide it with a complete air-conditioning system. This

was done early this summer when the building was enlarged.

Specifications called for a system limited to the minimum capacity and cost, and a number of interesting arrangements were worked out in its design. For instance, by employing a ventilated roof space, heat gain through the roof was cut down one-half, reducing the amount of refrigeration required. J. N. Livermore and P. S. Hosman explain the air conditioning in their article on page 593; it's full of ideas you'll find helpful.

• "Piping Steam for Heating Oil Tanks" is an article that's the result of some real study on the part of its author, Samuel R. Lewis. It contains numerous diagrams showing various methods of heating tanks so that oil may be handled quickly and economically and tells what to watch out for. This paper starts on page 610.

• Don't miss "On the Job" this month. There's an article by E. C. Parker telling how the cost of extras was cut on one job where it was necessary to supply 70-F water to photo-finishing machines; an article by John J. Harman on the proper application of pipe wrenches to prevent damage to pipe; a chart in G. L. Davis' article shows how important clean boilers are to economy; and W. H. Wilson shows that safety should be considered in piping, supporting his point by two examples from his own experience.

• When radiators over-heat, fuel is wasted; as forced-circulation hot-water heating systems do not tend to equalize themselves (as does a gravity system) it is important that means be provided so that each

unit of radiation will receive its proportional share of water.

Erwin L. Weber uses thermocouples to test temperature drop through radiators when he balances a system. As he points out in his article on page 608, the cost of checking up a system by this means is money earned, as he has often found radiators wasting as much as $1.35 annually. On a big job and over a period of years this runs into real money.

• When the West Virginia Pulp and Paper Company decided to use high-pressure, high-temperature steam a few months ago it became necessary to erect a large overhead line to convey it 920 feet. After H. C. Schramm—who writes the article on page 598 describing the erection of the line—had spent quite a bit of time in a search of technical literature and obtaining experience with shielded electrodes, it was decided to arc-weld the line using such electrodes. Mr. Schramm tells just how this piping was put up and how it was welded—and he doesn't neglect those points every industrial piping man will want to know about.

• Much attention has been given of late to chimney action in tall buildings, so Joseph W. Degen of the Equitable office building in New York City made tests last winter to find out just what effect flue action had on the heating of his building. He describes his results in the article starting on page 602. This is one of the first papers ever published presenting actual "operating data" on this subject.

• Two curves that look just about the same are shown in the chart in the short article "Does Cooling Pay?" One of them is a plot of the outside temperature and the other shows the actual daily re-

ceipts of an air-conditioned restaurant. When the outdoor temperature goes up, the public seeks a comfortable place to eat and the restaurant that can supply that comfort reaps a worthwhile reward. Daily receipts on some hot days were about double those on comfortable days. The chart answers the title's question as regards this restaurant in no uncertain terms.

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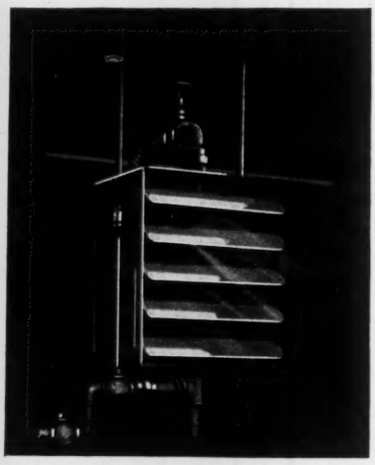

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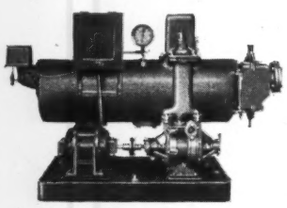
SIMULTANEOUS CAPACITY!

That quality of being able to do two things at once. A trap drummer can do it, but not all heating pumps can. A return line heating pump has to pump air and it has to pump water. Will the pump you contemplate buying pump both at once? A square foot rating on a heating pump represents nothing but the *opinion* of the manufacturer of that pump that it will handle the returns from a heating system of a certain size. The important thing is: How much air and water will it pump in a minute? After all, that is what you are buying, isn't it? So much air and water a minute pumped at a definite horse power.

Now most manufacturers of heating pumps do give a definite rating of water and air, and most pumps will pump their rated volume of air *or* of water. But what it is important for you to know is whether the pump will handle both together, at once, at the same moment. In other words, *SIMULTANEOUSLY*. Understand, not "Will it pump air and water at the same time?" Some pumps will even do that. Will it pump its *FULL RATED CAPACITY* of air *AND* its *FULL RATED VOLUME* of water and at its *RATED HORSE POWER*, and do it *SIMULTANEOUSLY?* The answer to that simple little question means a perfectly operating heating system with ample reserve capacity for zero weather, and it means a big saving in electricity. It means dollars. Make sure you have it right when you specify a pump, and you will specify a Jennings.



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Jennings Pumps

SIMULTANEOUS CAPACITY

capacity and cost had to be limited to bare necessity in AIR CONDITIONING this Detroit Edison Building

By J. N. Livermore* and P. S. Hosman*

Here is a building—a research laboratory—in an industrial location in which the atmosphere is heavily dust-laden. To carry on the investigations and tests for which the building is used, cleanliness is a requisite and windows and doors had to be kept closed. Keeping them closed meant unbearable conditions, particularly in the hot summer months. So a system which would give year 'round control of air conditions was called for—specifications were for a job in which capacity and cost would be limited to bare necessity.

This article describes in detail how a complete air-conditioning system to do the job was designed and installed—which, to meet the requirements for economy, had to solve the problems of:

- **. . . Arriving at the maximum cooling load.**
- **. . . Reducing heat gain through roof.**
- **. . . Piping water to make one pump serve for two.**
- **. . . Connecting refrigeration equipment for both high and low-temperature work.**
- **. . . Centralizing thermostatic control.**

The system has been in continuous operation since April 1, 1932.

*Mechanical Engineer, Drafting and Surveying Bureau, The Detroit Edison Company.

THE Detroit Edison Company has for some years maintained a research laboratory situated on the same property with its Delray Power House and Waterman Switching Station. The atmosphere in the locality of this building is very dirty as it is heavily laden with dust from nearby coal piles, and with an extra burden of dirt from a surrounding manufacturing district. In the laboratory, investigations and tests of every description are conducted, varying from that of finding the dielectric loss of underground cable insulation to determining the wearing qualities of boiler tenders' overalls.

The special demands on the laboratory have constantly increased, and consequently the number of delicate instruments and refined analyses have increased. For much of such work cleanliness is a prime requisite and is paramount for all. Closing the windows to keep out dirt accomplished this result but in the hot summer months this procedure made conditions unbearable. The chemistry laboratory, which should be isolated from the rest of the building at all times, found itself dependent on window ventilation, both winter and summer, unless doors to adjoining rooms were opened with the resultant passage of undesirable odors into the balance of the building.

This state of things confronted the research de-

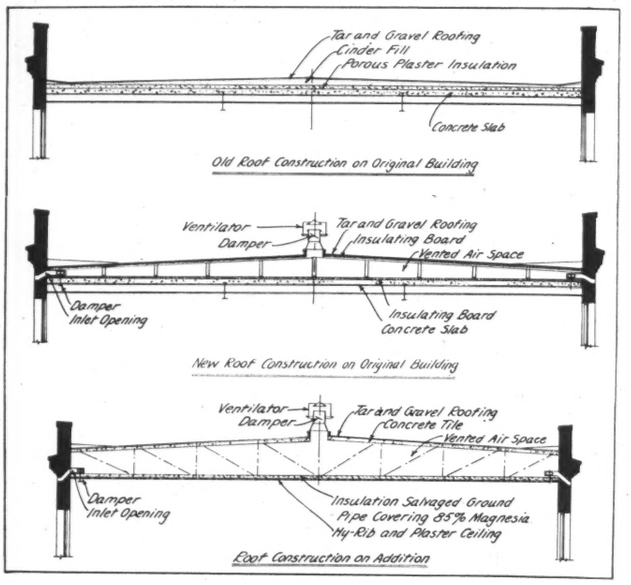
partment when the construction bureau of the company was asked to design and install an air-conditioning system. An addition was about to be erected at the same time, which would increase the building's volume approximately 65 per cent, and a system which would give year 'round control of air conditions in the entire structure was requested. Primarily the installation was to enable the occupants to keep all doors and windows closed without causing discomfort. The management stated that it was not essential, during the hottest summer days, to provide a degree of comfort which materially excelled that prevailing outside; this was pointed out with the idea of limiting the capacity and cost of the job to bare necessity.

Refrigeration Necessary to Meet Requirements

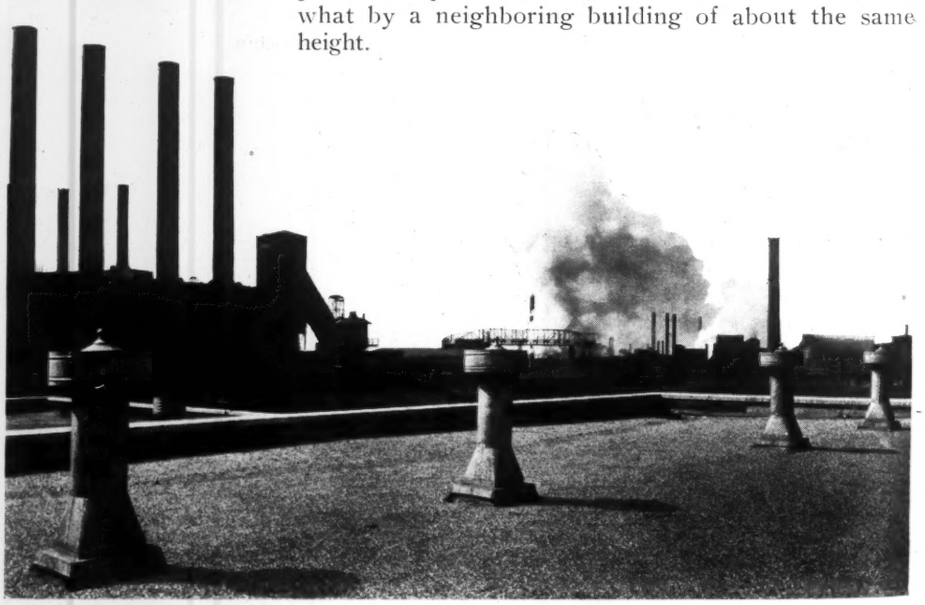
It was suggested that the building be ventilated by circulating quantities of filtered outdoor air, the volume being varied to suit the cooling or aerating requirements. The conclusion was soon reached, however, that due to the large amount of inside heat to be absorbed in summer months, this scheme would require too great an amount of air to keep the temperature rise within reasonable limits, and would result in drafts and unwieldy duct sizes. It would not, therefore, be possible to meet, or even approach, the ventilating requirements without the aid of refrigeration.

Determining Maximum Cooling Load

To interpret definitely the results to be obtained, effective temperature charts were used. Maximum cooling load was assumed to occur on a day when the dry-bulb temperature was 95 F, the wet bulb 75 F and the wind velocity 300 f.p.m., an outdoor effective temperature of 80 F. For purposes of design it was assumed that the corresponding indoor effective temperature would be held 3 F lower or 77 F. The 77-F effective temperature would be produced by maintaining 85-F dry bulb, a 71-F wet bulb and an air movement of 25 f.p.m. Designing for this improvement of comfort over that prevailing outside gave a desirable margin of plant capacity for unusual weather conditions. Whether or not greater comfort was maintained inside than out would be largely a matter of operation rather than design, in any event.



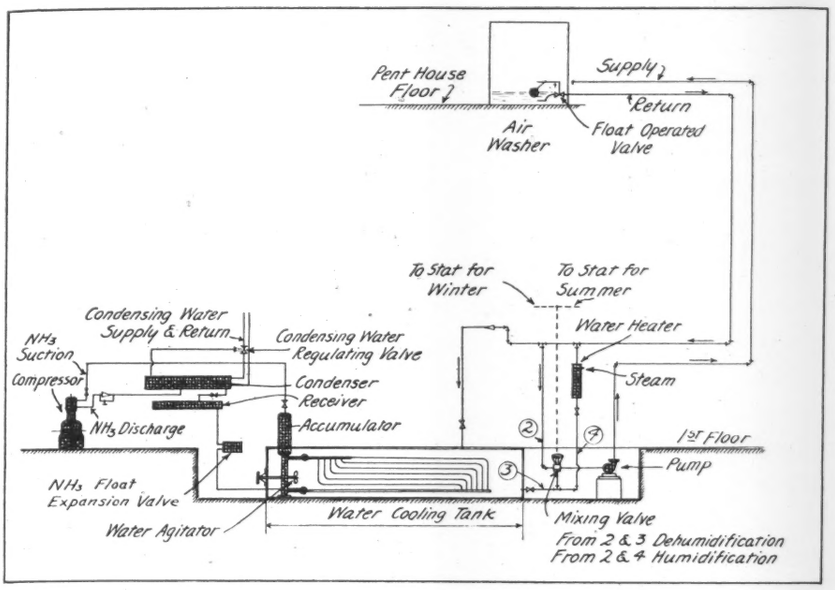
By using a vented roof space as shown in the drawing above, heat gain through the roof was reduced over one-half and the refrigeration required was cut down to economic advantage. The view below shows the roof ventilators and indicates the industrial district in which the building described is located.



The building under consideration is a three-story brick structure with concrete floors, L-shaped in plan and without a basement. The window sash are steel, double ventilated. All sides of the building are exposed except the south, which is sheltered somewhat by a neighboring building of about the same height.

The prevailing winds in this vicinity are westerly. To determine the cooling load, each of the rooms was studied separately. Special consideration was given many of the rooms, such as the chemistry laboratory, which was to be isolated and from which there was to be no recirculation. The room containing the motor-generator and synchronous converter sets was to be cooled by natural ventilation supplemented by mechanical ventilation later if found necessary.

A summary of loads anticipated from the various rooms was tabulated as to time (i. e., morning, afternoon and evening) with regard to the heat of the



The scheme of water and refrigerant piping is shown here. The mixing valve is supplied from Lines 2 and 3 for dehumidification and from Lines 2 and 4 for humidification during winter.

sun through glass and other building materials, and to the use of electric lights. By this means, the maximum refrigerating load was found, and was a basis for selecting a 40-ton ice machine.

Reduce Cooling Load by Venting Roof

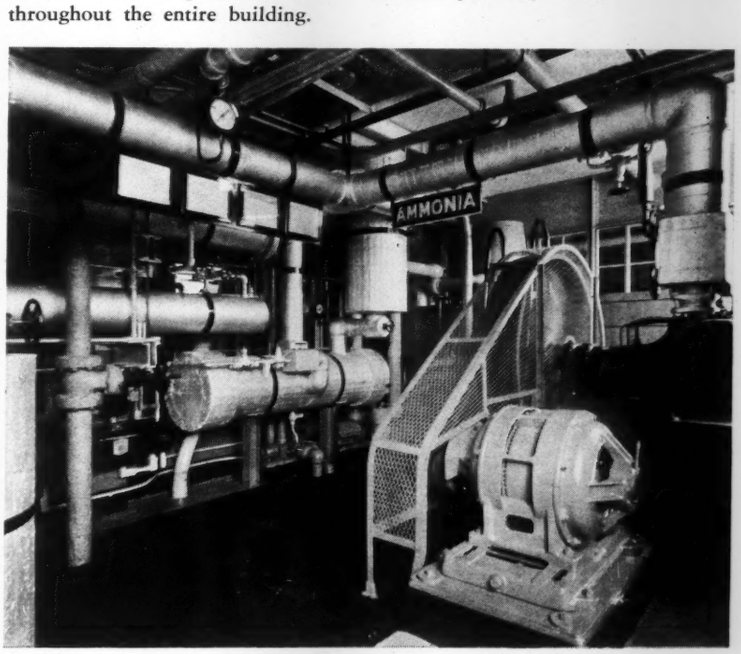
The cooling required for the rooms on the third floor was found to be greatly affected by the roof. The existing building was sheltered by a roof consisting of a 4-in. concrete slab, 4 in. of porous insulating plaster topped with cinder fills of varying thickness and two-ply paper roofing. Moisture had collected in the cinder fills causing them to give off a gas which blistered the roofing to a point beyond repair. At the same time, the insulation beneath had been reduced to a wet muck of doubtful insulating value. It was planned to clean the roof down to the slab and re-cover it. Before doing this, however, considerable study was made to find a roof covering better suited to warm-weather insulation.

The roof surface temperature was assumed to reach 160 F due to solar radiation, and the ceiling temperature inside assumed at 90 F. This temperature maintained throughout a day would soak through the materials giving a temperature difference of 70 F acting as pure conductance in the late afternoon. The overall coefficient of conductance for the existing roof (in good condition) is 0.10 Btu / Sq. ft. / 1F / Hr. and with the above assumptions 7.0 Btu/Sq. ft./Hr. heat inflow might be expected.

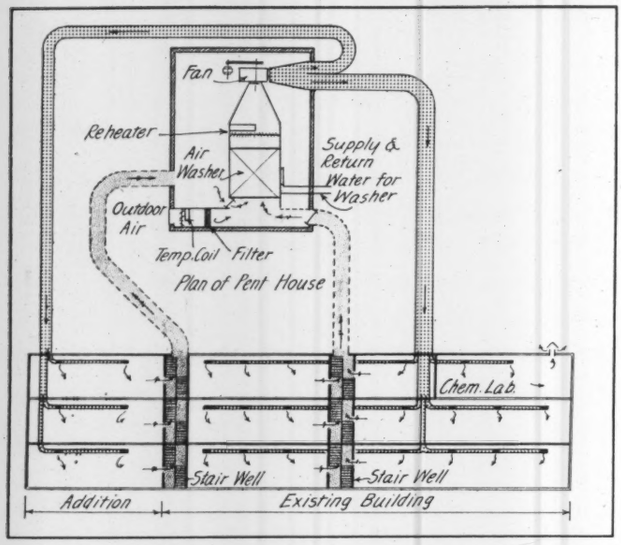
Turning away from the more usual types of insulation, a construction feature common in cold storage buildings suggested itself; this consists of a ventilated air space above the ceiling of the top floor. Assuming that the mean air temperature might reach 105 F in this space on a 95-F day, the heat inflow through the ceiling would be that corresponding to a 15-F temperature difference through the old concrete slab insulated with fibrous board. The overall coefficient of conductance for this slab being 0.226, the heat inflow would be 3.39 Btu/Sq. ft./Hr., a considerable improvement over the existing roof in summer.

By using a vented roof space on both the old building and the addition, the maximum cooling load is reduced approximately 32,400 Btu/Hr.[1] or 2.70 tons of refrigerating capacity. With the vent openings on the roof closed in winter, the overall coefficient of conductance for the upper roof and ceiling slab below is substantially the same as that of the old roof during the heating season. Estimates of cost show that the additional amount required to provide this type of construction rather than replace the old covering was well justified.

[1] In view of data presented in "Heat Transmission as Influenced by Heat Capacity, and Solar Radiation," by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (presented at the 1932 semi-annual meeting of the *A. S. H. V. E.* and published in HEATING, PIPING AND AIR CONDITIONING, April, 1932) the refrigeration economy should be even greater than that anticipated. (J. N. L.)



The ice-machine room, with the brine tank appearing across the far end. The cold-water tank is under the checkered plate floor. This installation was so designed as to enable the refrigerating equipment to be used for low-temperature experimental work when the air-cooling equipment is not in operation. A remote-reading resistance type thermometer located in the ice-machine room enables determination of temperatures throughout the entire building.



The scheme of air circulation is diagrammed in this drawing. Note that the duct system is divided into two zones, enabling better control with varying sun effect on the different exposures.

Central Cooling System Used

A central system of cooling seemed a logical arrangement for this job. Two stair wells, one at the center of the old building and one adjacent to the addition, were chosen for use as shafts for recirculated air, and the location of these determined the location of the fan pent house between them on the roof. In the pent house, recirculated and outdoor air are drawn through the air washer and discharged into the duct system. Outdoor air is drawn into the circuit in an amount sufficient to make up for that vented out of the chemistry laboratory and toilet rooms with enough surplus to keep the building under a slight pressure. All outdoor air is drawn through paper filters before entering the washer.

The winter function of the air-conditioning equipment is limited to air renewal and to the control of humidity as the entire building is heated by direct radiation. To accomplish this, the fan is run at a low speed supplying air at a constant temperature and moisture content.

"Zoned" Ducts Aid Operation

The duct system handles a maximum of 26,000 c.f.m. in summer. The ducts were designed to give an air velocity of 800 f.p.m. and the throat approaching each diffuser to increase the outlet velocity to 1,000 f.p.m. This high exit velocity prevents cascading of cold air directly to areas beneath the outlets, and by jetting the air at high elevation for a distance, good diffusion with the warmer room air is obtained. Care was exercised to avoid columns, low beams and room fixtures which might deflect the air jets downward. Sweep sheets and guide vanes are liberally used throughout the system to assure good distribution between the outlets, and over the face of each outlet. Each outlet is equipped with a damper to permit balancing the system.

The duct system is divided into halves, one cooling the east rooms and the other the west. With the aid of two dampers in the pent house, it is possible for an operator to vary the volume delivered to either side of the building to suit the varying cooling demand from change of sun exposure during the day.

Installing Ducts No Easy Job

Those who have installed extensive duct systems in existing buildings will readily appreciate the difficulties which are met with and the numerous compromises with the ideal which must be made. This job was no exception. The laboratory had been built to provide facilities for many sorts of test purposes wherever one might choose to make a setup. Duct runs seemed to be blocked in all directions by lighting, power or spare conduit, by high-pressure, low-pressure or return steam lines, by compressed air or vacuum piping, by city gas, city water, drain or brine lines, or by some expensive test setup which could not be disturbed.

How One Pump Does Work of Two

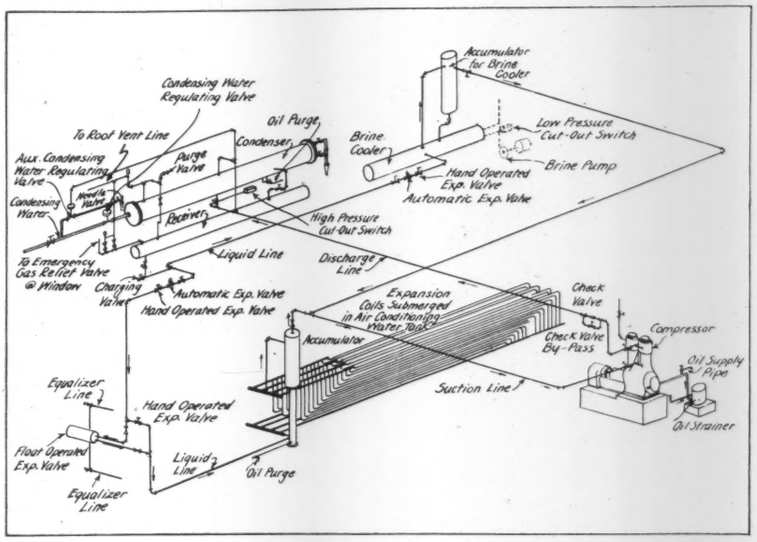
An accompanying diagram shows the scheme of water circulation used in both winter and summer. The circulating pump is located in the cooling-tank pit at the first floor and supplies the air washer in the pent house. Spray-water temperature in winter is regulated by a dewpoint thermostat controlling a mixing valve which, in turn, proportions the amount of air-washer return water passed through and bypassed around an enclosed steam heater. The mixture then supplies the pump suction. In winter the cold-water tank shown is cut out of the pumping circuit and the pump works on balanced suction and discharge legs. Any scheme which would eliminate this long pumping distance, would involve a second pump and mixing valve located next to the air washer, and it was found that the investment costs of the extra equipment would exceed the cost of power wasted in pipe friction in the scheme shown.

In summer, the temperature of the spray water is controlled by mixing a portion of the water returned from the air washer with the water withdrawn from the cold-water tank. This means that the return water line had to be designed not as a simple overflow pipe, but as a line which maintains a solid stream of water free from entrained air, feeding both the pump suction and the cold-water tank. This was accomplished by locating a float valve on the entrance to the return line which serves to throttle the flow just enough to maintain a proper level in the air-washer tank and to keep the entrance to the return pipe submerged.

50-hp. Motor Drives Compressor Through V-Belts

The ammonia compressor used is designed to operate with flooded cooling surface and to operate automatically, maintaining water in a tank between 35 and 40 F by thermostatic control. It is a 7½-in. by 7½-in., vertical, twin cylinder, single-acting type, driven through V-belts by a 50-hp. motor. The condenser is a multi-pass, shell and tube type. With the compressor operating at 275 r.p.m., and the water in the tank at 40 F, the plant has a capacity of 40 tons.

In order to reduce building vibration caused by the equipment, much of the machinery is set on vibration-absorbing bases. The ammonia compressor and motor and the supply fan and motor, are placed on live rubber sheets 1-in. thick, perforated with 1-in. holes. The holding-down bolts are arranged so that there is no direct metal-to-concrete connection between the machine and its base.



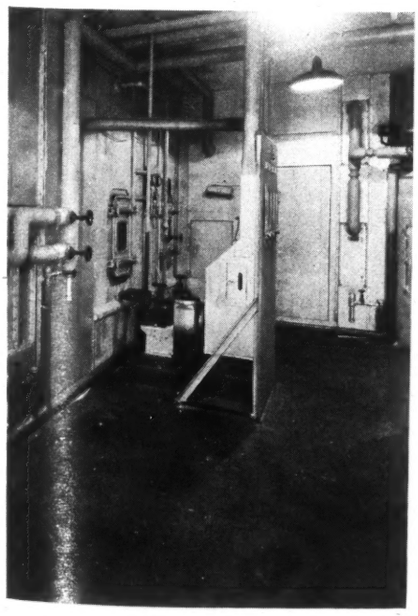
An isometric drawing of the refrigerating equipment.

Cold Water Tank Carefully Insulated

To economize space, an abandoned furnace gas flue beneath the first floor was widened sufficiently to contain the cold-water tank, which is 5 ft. x 26 ft. x 3 ft. 3-in. deep, and made of welded 3/16-in. steel plate insulated with 4-in. cork. The sides of the tank were covered with 2-in. of slab cork before lowering it into the pit; the remaining 2-in. consists of granulated cork which was rammed hard between the tank and the pit walls and sealed at the top with asphalt. Submerged at one side of the tank are the ammonia expansion coils with a vertical steelplate baffle running their full length. This baffle forms a channel through which an agitator circulates water in a stream confined close to the cooling surface. In the pit at one end of the tank is located an ammonia float valve which serves as an expansion valve and keeps the coils flooded with liquid ammonia. The compressor withdraws the evaporated refrigerant at a rate which maintains 40-lb. pressure in the coils, giving a corresponding boiling temperature of 25.8 F.

Keeps Slugs of Ammonia from Suction Line

The gas and liquid headers at the top and bottom terminals of the expansion coils are connected by a large vertical pipe which also leads to the bottom of the suction accumulator. This arrangement causes any liquid entrained in the evaporated refrigerant to drain directly back to the lower portion of the coils. As the compressor comes up to a constant full speed immediately on starting, without manipulation of the suction valve, this provision is important in avoiding the possibility of slugs of liquid ammonia entering the suction line.



The cabinet in which control for the system is centralized to the benefit of the cost of operation is shown in this view of the penthouse fan room.

Refrigeration Plant Serves Double Purpose

During seven months of the year the refrigerating plant is not needed for air conditioning, so it is connected for alternate use in experimental work. On the second floor of the building is a small, heavily insulated room, formerly used for high temperature work, which is cooled to temperatures as low as 0 F. This was arranged for by installing two banks of brine coils in the cold room, and a shell-and-tube brine cooler, brine tank, and brine circulating pump in the compressor room.

Referring to the diagram at the top of this page, it is seen that the water-cooling coils and the brine cooler are in parallel and must operate at the same evaporating pressure and temperature if used simultaneously. However, during periods when the building does not require cooling, the automatic expansion valve on the brine cooler may be set to produce brine temperatures as low as —10 F. When operating under these conditions, valves are closed so as to cut the cold-water cooling coils out of the ammonia circuit.

Control Equipment Is Centralized

All thermostats are grouped in a partitioned cabinet, with a thermostat, a thermometer, and a pressure gage behind a glass door in each compartment. A 4-in. sampling pipe which conveys air from a point where its state is to be controlled runs to the top of each compartment. At the bottom of the control cabinet a small exhauster is located which draws air through all the sampling pipes and compartments into a common suction box, from which it discharges to the room. This device centralizes all temperature control at one point.

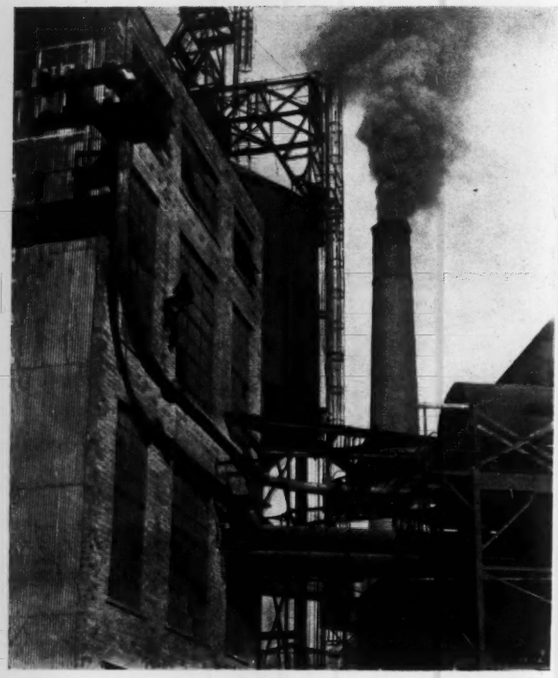


Fig. 1—Note the construction platform on the pipe in this view. How the flanged joints were assembled is described fully in this article.

By H. C. Schramm*

Erecting a High-Pressure, High-Temperature Steam Line

at West Va. Pulp & Paper Co.

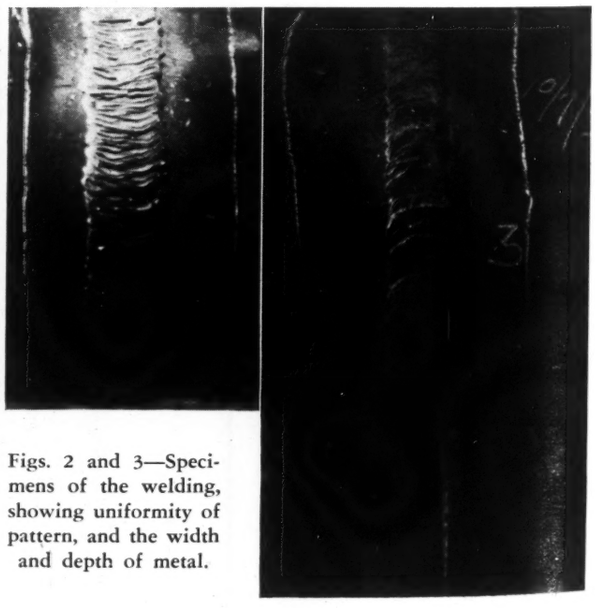
PREVIOUSLY to July, 1930, the steam requirements of the Piedmont plant of the West Virginia Pulp and Paper Company were entirely supplied with steam at 180-lb. gage pressure. Since March, 1931, a part of the 180-lb. steam has been replaced with steam at a pressure of 600 lb. and a total temperature of 720 F. A pipe line 920 ft. long was required to convey this steam from a boiler to a turbine exhausting at 20 lb. and inducing or extracting at 180 lb. Desuperheated steam through a temporary line was used while the new line was being erected.

The pipe used in this line is 14-in. O. D. seamless hot-drawn tubing 21/32 in. thick. The Van Stone laps are the full thickness of the tubing and have a phonographic finish. The flanges are 23¾ in. O. D., regular 600-lb. standard forged steel, Van Stone, with high hub and square corners.

Arc Welding With Shielded Electrodes

After the writer had spent some two weeks "pushing" bare and coated electrodes in a welding school in order to become familiar with them, and had searched for references to their use for field welding a 600-lb., 720-F line, it was decided to arc weld the line using shielded electrodes. These could be expected to deposit metal equal to rolled steel, it was learned.

The general instructions given our men for this



Figs. 2 and 3—Specimens of the welding, showing uniformity of pattern, and the width and depth of metal.

*Engineer at Piedmont, W. Va., plant of the West Virginia Pulp and Paper Co.

That care in design and erection means economy in maintenance is demonstrated by this 600 lb.—720 F steam line, which was arc welded, using shielded electrodes.

A number of practical helps were worked out in the erection and are described by the author in this article:

Chill Bands Aid Welding
Instructions for Welding
Clamp Kept Pipe Aligned
Assembling the Flanged Joints
Removing Mill Scale from Pipe
100-Ft. Length Raised at One Time
Testing the Line and Putting It in Operation

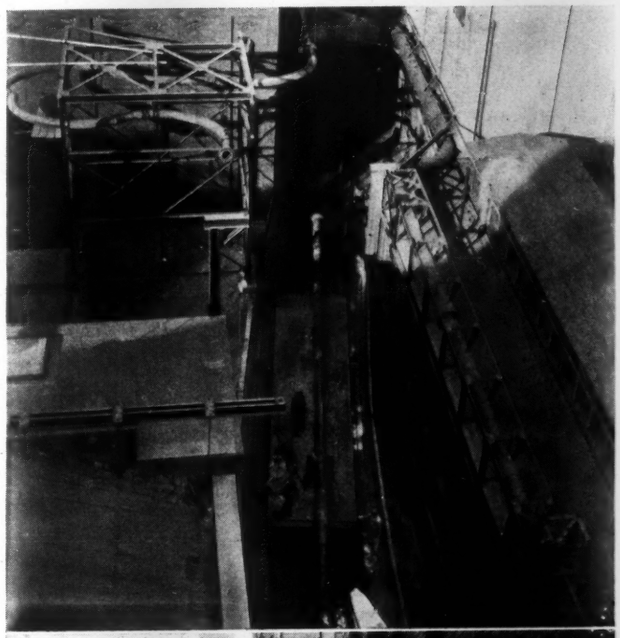
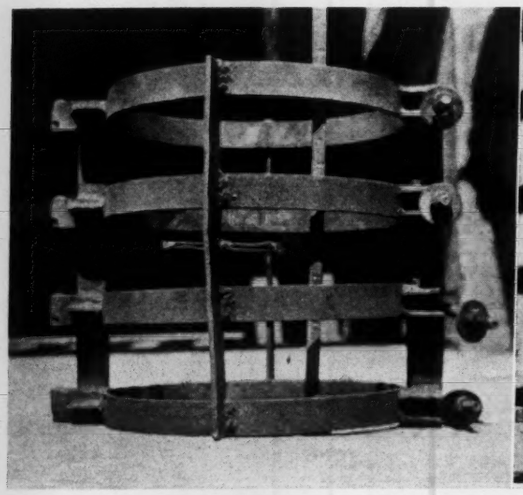


Fig. 4 (Top)—Longest piece of pipe erected in one piece on flat car. Fig. 5 (Middle)—Same pipe leaving car; note deflection. Fig. 6—(Bottom)—Almost landed. Opportunity for using the 25-ton locomotive crane in erection influenced location of the line and reduced the cost.

particular job were: The success of any weld depends largely upon the foundation. In pipe welding only a single V can be used. The edges should have left a shoulder about 1/16 in. high at the bottom of the scarfed edge, if the metal is more than ⅜ in. thick. There must be sufficient space between the parts that are to be welded together to enable the welder to obtain penetration and fusion at the bottom of the V. For metal more than ¼ in. thick this spacing is usually ⅛ to 3/16 in. Use the 3/16 in. spacing.

Place weld in three or four layers. Use 3/16-in. diameter rod for burning in first bead and larger rod, ¼ to ⅜ in. diameter, for the other layers. Weld first bead in bottom by alternate method, i. e., burn in four beads each 2 in. long on 90 deg. centers. Then fill in between these. The other layers may be continuous welds. Clean between each layer with sand blast. If any spots look doubtful, chip them out before proceeding with the weld. Watch penetration closely and adjust current when needed. Keep work clean at all times; the importance of clean work cannot be stressed too strongly. A vertical position weld will be impossible with this electrode if work is not kept clean. Write your initials with the electrode alongside of each weld you make so the responsibility for each weld can be definitely fixed and one man will not be held to account for any failure in the other fellow's work.

Figs. 2 and 3 illustrate the general appearance of metal deposited by each



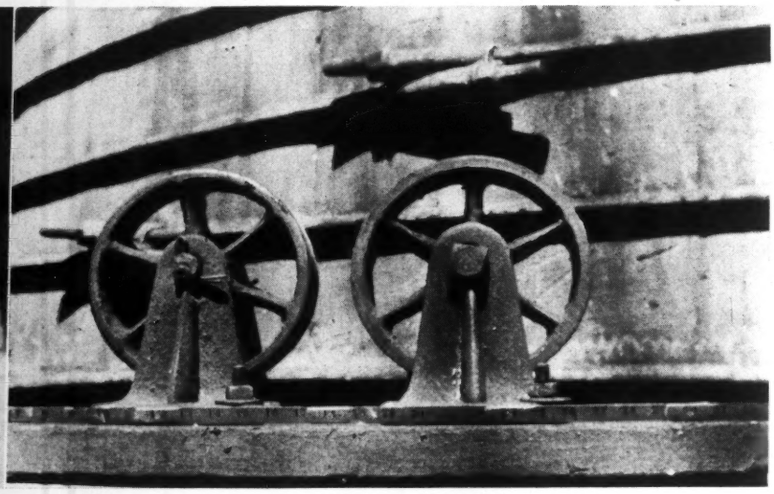


Fig. 7 (Left)—Pipe clamp devised to keep pipe in alignment during welding. Better ones are now on the market, says the author. Fig. 8 (Right)—Roller used to support pipe so it could be turned while welding.

operator. These specimens show the class of work done shortly after starting the job; quality of work improved as the job progressed. Fig. 3 shows the initial of the operator making the weld, and also part of a supporting roller.

Removing Mill Scale from Pipe

A special tool developed by the leader of the crew that erected the 14-in. pipe was used for knocking mill scale loose while the pipe was on the ground. Water-driven turbines such as are used for boring tubes in boilers were utilized for this. On the shaft of such a unit a steel disk replaced the cutter head. There were three holes in the disk. About a foot of ¾-in. link steel chain was bolted to the disk at each of these holes. The turbine was then run through each pipe and bend. Mill scale has not given any trouble.

Chill Bands Aided Welding

On less particular jobs we have found free spatters of weld metal, short pieces of welding rods and "icicles" of weld metal inside of pipes. Even thorough blowing with 180-lb. steam does not always remove all of this foreign material. A thin chill band of steel placed in a pipe will seal off the joint and prevent such troubles.

The chill bands were turned out of a piece of pipe. They were 2 in. wide and ⅛ in. thick; the outside diameter was 12⅝ in. A 1/16 in. wide by ¼ in. high fin was left on the outside of the band, located 1 in. from either edge. A chill band was used in each joint, the up-stream edge being welded to the inside of the pipe. The next length of pipe was fitted over the projecting edge of the chill band and the whole clamped together for welding.

The thin projection or fin served as an electrical contact point for striking the arc and prevented burning through the band itself, materially assisting in securing the desired penetration and fusion at the base of the V. Due to the fluidity of the shielded rod, the chill bands proved even more valuable than we anticipated.

Clamp Kept Pipe Aligned

Another detail solved on this job was how to keep two lengths of pipe in alignment and properly spaced while making a rolling weld, and for this the special clamp shown in Fig. 7 was designed and built. The 3/16-in. thick spacers were located on 90 deg. centers. The clamp had to have sufficient strength and rigidity to keep the pipes in alignment and at the same time not interfere with the operator. There is now on the market a clamp that is very much better than the one we made.

Rollers or dollys were used to support the straight pipes while making rolled welds on the ground; the design is shown in Fig. 8. Unfortunately pictures are not available showing these rollers in position supporting a section of 14-in. pipe, but the photo indicates the method of supporting and rolling the pipes. The sections were aligned by stretching a string along each side of the pipe; when the pipe was touching or equally distant from both strings, the section on the rollers was considered in alignment. The same type of roller mounted on two channels was used to support those straight sections that had to be welded in the air and that could be rolled.

Assembling the Flanged Joints

Steel "gasket" rings were used in assembling the flanged joints. A ¼ in. thick steel ring with a 1/16 in. wide projection on each side of the inside diameter placed between the Van Stone pipe ends formed an excellent base for welding. The projection centered the ring in the pipe. The flanged joints were assembled with the edge of the ring ¼ in. below the flange rim, and the ¼ in. space was filled with the first bead. Then the metal was built up ¼ in. above the edge of the flange by laying on two more beads.

The bolts were all pulled up just as tight as if a gasket were being depended upon to hold the pres-

sure. Every other bolt was then removed and as much of the welding done as free space permitted. These bolts were replaced and the other half of the bolts removed so the welding could be completed.

100-Ft. Length Raised at One Time

Along the boiler house a 64-ft. length of pipe, welded together, was raised at one time. This looked like a rather long piece of pipe to raise all at once and there was some question about whether it should be reinforced with I-beams or channels while being hoisted. It was finally decided that had the pipe been manufactured in one length there would have been no question about raising it. The butt-welded joint was supposed to be stronger than the pipe, and if it would not support its own weight another type of joint might have to be considered.

The longest piece erected is shown in Fig. 4 on a flat car. In Fig. 5 it is shown leaving the car and by Fig. 6 it is almost in position. This section measured 100 ft. from flange to scarfed end and was assembled, i. e., a number of short lengths welded together, parallel to the siding, loaded on the flat car and moved into position by the 25-ton crane.

A section of pipe 177 ft. long between flanges was erected in three parts and two welds made by rolling the whole after erection. A 222-ft. section of pipe located between towers was supported and rolled on five hanging rollers while making two rolled welds. These were spot-welded and then turned as when welding pieces together on the ground.

Testing the Line

After the line had been erected and welded from the boiler house to the first expansion bend it was pumped full of cold water and the pressure was built up with a hand-operated single-plunger pump until a gage indicated 1500 lb. per sq. in. hydrostatic pressure. This pressure was maintained on the pipe while an inspection of each welded joint was made. Each joint was thoroughly hammered with a 2-lb. ball pein hammer and the pipe was found to be absolutely tight. The other section of the line was subjected to the same hydrostatic test with the same result.

Parallel to and just above the 14 in. line a 2½-in. water line was installed to bring water from the boiler feed pumps to a desuperheater in the power house. The operating temperature of the water is 215 F at 600 lb. gage pressure. The pipe is 2½ in. extra heavy and was ordered in random mill lengths. It was welded with the same shielded rod, tested and found tight.

Putting the Line in Service

The turbine was started March 11, 1931, using 180-lb. steam to turn it over. On the 16th it was ready for high-pressure steam. The entire line was then pumped full of water for a 1500-lb. hydrostatic test, and this pressure left for twelve hours. No leaks being found the line was drained and then very thoroughly blown out with 180-lb. steam. The pipe was next brought up to 180-lb. pressure at 420 F temperature and this pressure maintained over night. The following morning the pressure was blown down and the pipe connected to the high-pressure boiler. The boiler was brought up to pressure and the turbine started on 550-lb., 720-F steam.

When the high-pressure boiler is out of service the 14 in. line is kept warm with 180-lb., 420-F steam. This has happened about 75 times since March, 1931; pressure has been entirely off the 14 in. line three times.

There were fifty-eight welded joints altogether; twenty flanged joints, two sleeves on two expansion bends, two position welds on pipes, and two heads inside of fittings at dead ends totaled twenty-eight position welds. The remaining thirty were rolled welds, five of which were done with the pipe on hanging rollers. During construction of the line there was ordered 300, 475, 200 and 650 lb. of ⅛, ¼, 3/16 and 5/32 in. diameter shielded rods. These were partly used for practice welding in the shop, for practice welding on several other jobs, for actual work on the 14-in. line, for work on a 2½-in. water line and some left for stock. There is no record available of the actual pounds of each size used on the 14-in. steam line.

Work on this pipe line was started in September and finished in November. The two welders and erecting crew did not work on it all the time as there were other construction and repair welds that had to be done by these men. The dates, therefore, do not measure the actual construction time. Itemized costs of erecting the line were not kept separate from the other jobs; hence they cannot be given.

Enough time was spent on each joint to make what we considered a first-class weld. There has been absolutely no trouble with the pipe line.

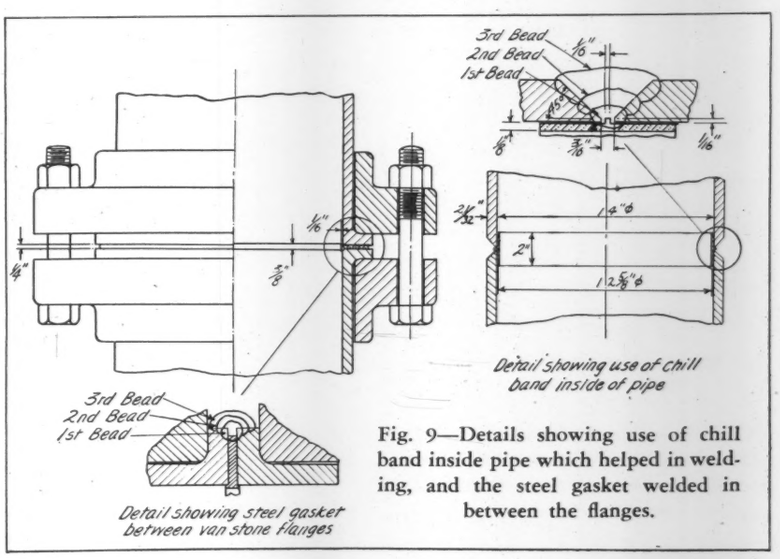


Fig. 9—Details showing use of chill band inside pipe which helped in welding, and the steel gasket welded in between the flanges.

Heating the Tall Building

Does "Chimney Action" Overcome Exposure in Upper Floors? . . . *By* Joseph W. Degen*

Chimney action in a tall building has a pronounced effect on the design and operation of the heating system. The study made by the author in the 40-story Equitable Building (above) is one of the first to give actual data on chimney action; the results are presented here.

This study leads to these conclusions:

Seemingly more radiation is needed in the lower floors of tall buildings than in the upper floors.

Since the inleakage and, consequently, the cooling of the lower floors is occurring twenty-four hours a day in cold weather, there should be some method of horizontal zoning of the heating system whereby the lower floors can be independently heated. By this means the average temperature can be kept more nearly the same for the upper and lower floors.

The up-flow of warm air tends to counteract the effect of a north exposure, thus perhaps making unnecessary increased radiation for such a condition.

OF LATE there has been a noticeable increase in interest in the effects of "chimney action" in tall buildings. By chimney (or stack or flue) effect or action of a tall building is meant the inrush of cold air in the lower floors of the building, which would correspond to the inrush of air into a furnace; its subsequent heating by the radiators, which would correspond to the rise in temperature of furnace gases; its natural upward flow as corresponds to flue gases in a chimney; and its exhalation at the upper floors similar to the escape of flue gases from the top of a chimney. As the draft of a chimney is effected by the differences in temperature (and, consequently, the specific weight) between the flue gases and the surrounding air, so the draft in a building is effected by the corresponding factors. The great difference is that this draft is desirable and planned for in a chimney, but is disadvantageous and combatted in a tall building.

The ill effects of chimney action are many, as are the ways in which it can render futile the efforts of a designer of the heating and ventilating system in a tall building.

As the number of tall buildings have increased and as the men who operate them began to relate their experiences there has grown a belief that this idea of increasing radiation due simply to upper-floor exposure is perhaps not always a necessity and is certainly not the whole story.

There is a growing appreciation of the effect of chimney action on the relative proportions of radiation between the upper and lower floors. As the cold winds were supposed to wash heat from the upper floors, the inrush of the cold outdoor air to the lower floors actually washes the heat from them and transfers it in some measure to the upper floors, so that in most cases instead of increasing the radiation in the upper floors to allow for exposure it would seem better to decrease it and increase the lower floor radiation correspondingly. The great difficulty is to decide just what the actual change should be and how it can be determined. This entails a study of present structures and their experiences.

Many suggestions as to means of counteracting chimney action have been offered. They comprise mainly in-

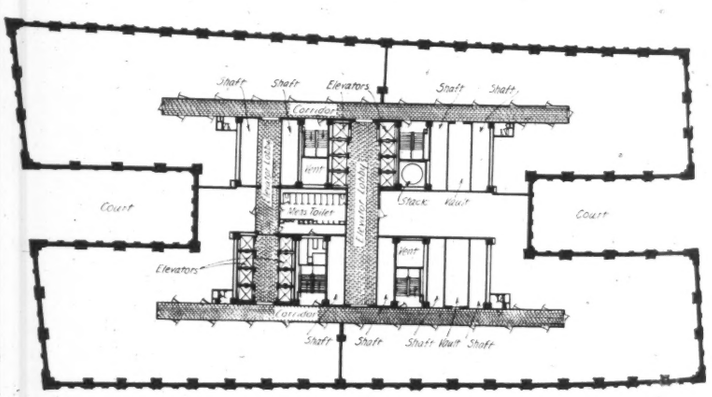


Fig. 1—Typical plan for the seventh to thirty-ninth floors.

*Supervising engineer, Equitable Office Building Corporation, New York City.

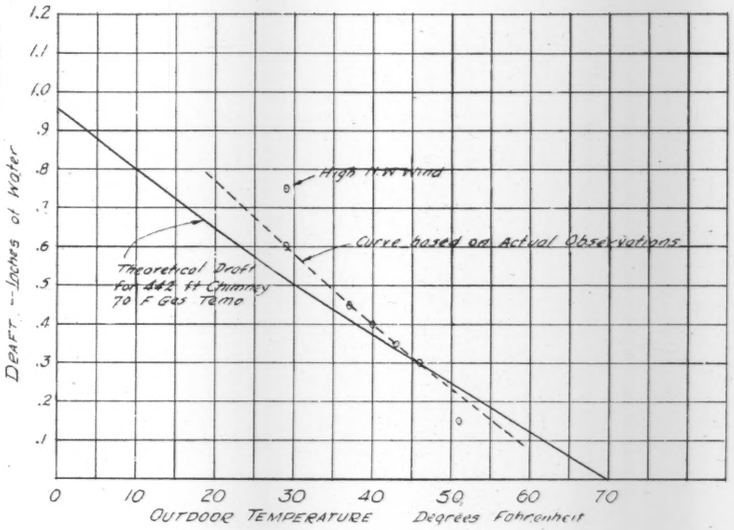


Fig. 3—Draft observed in one room by means of a U-tube and curve showing comparison with theoretical draft for a chimney 442 ft. high.

creased lower floor radiation; horizontal isolation by sealing of zones or floors, the creation of an artificially produced higher pressure zone in the upper stories; more efficient sealing and weather-stripping of lower floor windows, doors and shafts, etc.

Chimney Action Study Made

In order to be able to offer some actual and perhaps useful data on the subject there were several readings and a study made of chimney action in the Equitable Building, New York City. The building is forty stories high (542 feet) and occupies a full city block of 48,941 sq. ft. The six lower floors (1 to 6 inclusive) are full rectangles of floor space. From the seventh floor to the roof the shape is that of an "H" with an average of some 30,000 to 31,000 sq. ft. of net rentable area per floor, as shown in Fig. 1. The floor heights vary somewhat, with 12 ft. 2 in. as the most typical. The gross volume above the street level is 22,639,546 cu. ft.

With the exception of the areas below street level, the main floor and the two upper floors, the building is heated by 157,000 sq. ft. of exposed, non-recessed, cast-iron, manually-controlled radiation. Radiators are, in most cases, located one to a window. The basements, main floor and the two upper floors are heated by combination direct and indirect systems.

The first-floor direct radiation is fed through a separate upfeed system supplying this floor only. The other floors are fed through risers feeding up and down from a header on the thirty-sixth floor.

Fig. 2 shows the floor to floor distribution of the installed radiation from the second to thirty-seventh floors inclusive, together with the related figures of gross floor area, ceiling height and ratio of volume to radiation. This last ratio shows slight variation between the floors of "H" shape, that is from the seventh to thirty-seventh, which would indicate that no special allowance was made for the upper or lower floor radiation.

Exhaust steam from non-condensing Corliss steam engines in the power plant is the usual heating medium. For early morning warming-up it is necessary, in severe weather, to use live steam, governed as to time and rate of use by the outdoor temperature and temperatures on the main, second and third floors. These three lower floors are taken as governing points for temperature readings, first, because these floors are cooled off first and quickly by the inleakage of cold air due to the chimney action, and second, these lower floors are on the end of the lines and are the last to receive steam.

Draft-Gage Readings Show Pressure

During the winter of 1930-31 draft-gage readings were taken in Room 409, 442 ft. below the main roof of the building. An ordinary U-tube with water was used—a rubber hose being dropped out the window which was then closed as far as possible without closing the tube. These readings are plotted in Fig. 3. There is also plotted on the same ordinates the theoretical draft line of a 442-ft. chimney with a 70-F average flue-gas temperature. The two curves show a divergence above and below the temperature of about 46 F, which can be explained by the fact that at the lower temperatures the windows are all kept shut as tightly as possible, thus raising the average building temperature to perhaps 72 or 74 F, and at the higher temperatures, as a result of milder weather, more windows are

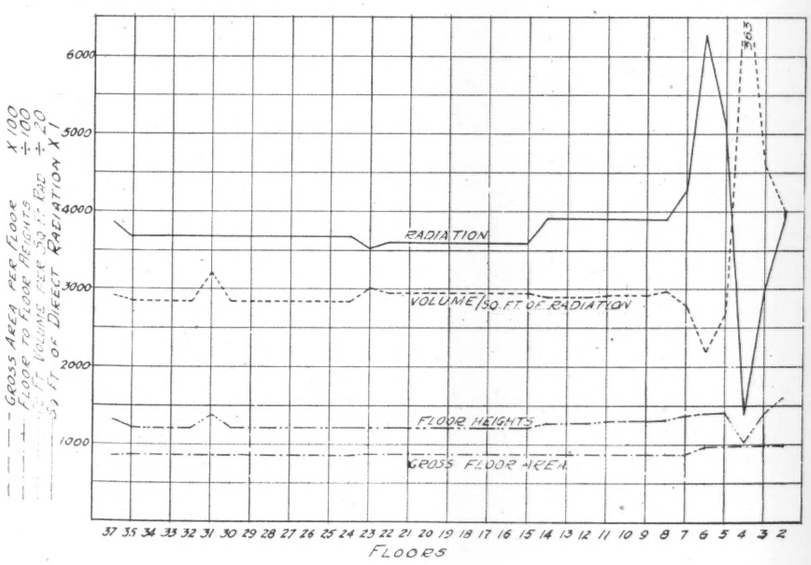

GROSS AREA PER FLOOR X 100
FLOOR TO FLOOR HEIGHTS X 100
CU. FT. VOLUME PER SQ. FT. FOR 50 FT. OF DIRECT RADIATION X 20
6000
5000
4000
3000
2000
1000
365
RADIATION
VOLUME/50 FT OF RADIATION
FLOOR HEIGHTS
GROSS FLOOR AREA
37 35 34 33 32 31 30 29 28 27 26 25 24 23 22 21 20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5 4 3 2
FLOORS

Fig. 2—Floor-to-floor distribution of the radiation, gross floor area, ceiling height, and ratio of volume to installed radiation.

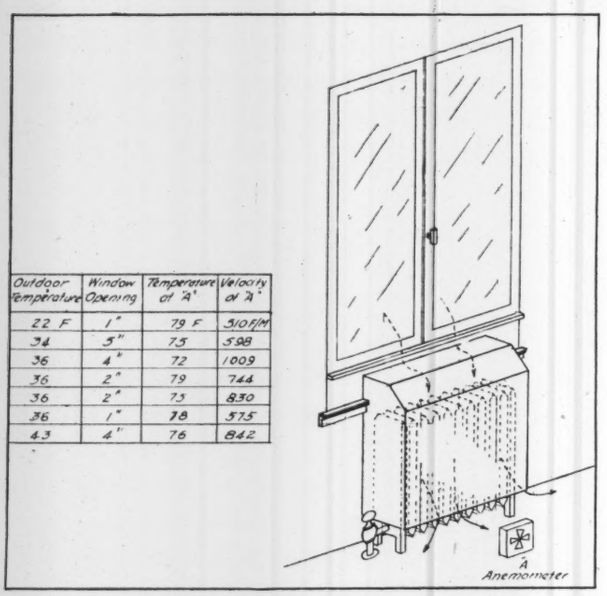


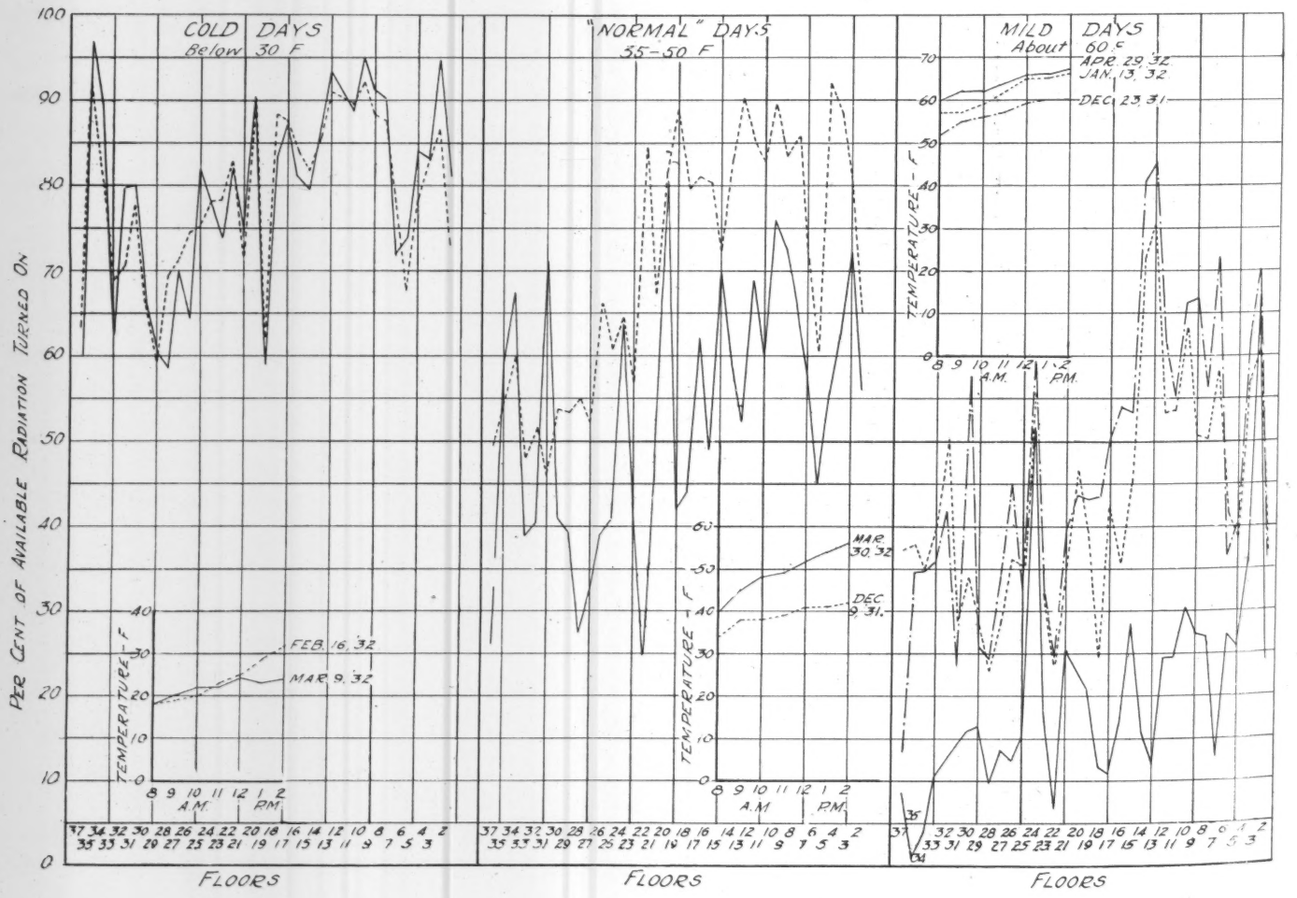
Fig. 4—Radiator enclosure designed to utilize difference in air pressures for ventilating an office room.

opened throughout the building, thus by the free circulation cutting down both the average temperature and draft.

Pressure Difference Used for Ventilation

An interesting attempt was made to use this difference between the outdoor and indoor air pressures by the construction of an inner casement window and radiator enclosure, as shown in Fig. 4. The idea was to open the regular window, which is of the balance type, shut the casement sash and guide the cold outdoor air over the warm radiator, thus supplying to the room an abundance of warm fresh air. The first installation was made on the fifth floor in an office where, due to the large number of occupants, the window could not be opened without a complaint from someone about an objectionable cold draft. The table shown with the sketch gives the results obtained, the velocities being especially notable. The objective of an "abundance" of fresh air was easily realized. The abnormally high floor draft drew complaints from those occupants who were situated at the inner wall of the offices where the air current was changed from the horizontal to vertical causing irritating discomfort in trouser legs. To overcome this the design was changed by the addition of a vertical baffle about three inches in front of the original enclosure, which resulted in a vertical flow of air, inducing a good circulation and leading to no complaints. Since then, some twenty of these enclosures have been installed in offices where the large numbers of occupants pre-

Figs. 5, 6 and 7—Results of count made on cold, normal and mild days to find amount of radiation turned on by tenants, indicating practical effect of chimney action on operation of heating system.



vented use of a normal amount of window ventilation.

Radiator Count Shows Effect of Chimney Action

In order to have some idea as to the practical effect of the chimney action on the heating system, it was decided to enlist the unsolicited and unconscious cooperation of those for whom the heating system is operated—the tenants. Regardless of the theoretically-obtained or scientifically-measured results, no heating system is satisfactory when it is the cause of complaints. The means taken to make use of the tenants' cooperation was to maintain normal, complaintless operation of the heating system and then count the radiators actually in use. Since all radiators (with the exception of some 25 out of 5000) are manually controlled, it was reasonable to assume that any radiator found on or off was so operated by the tenant.

Accordingly, during the winter of 1931-32, there were made seven counts of the number of radiators turned on and off from the second to the thirty-seventh floors inclusive, between 11 a. m. and noon on seven different days. The days selected for the counts were toward the middle of their respective weeks, in order to obviate the possible effect of unusual cooling of the building over a week-end. A stable normal condition was sought.

The results of these counts are plotted in Figs. 5, 6 and 7. The radiation found in use is expressed as a percentage of the total radiation on the floor. This percentage method is selected as best showing a comparable figure for floor-to-floor interpretation.

The general slope of these graphs from a relatively low percentage on the upper floors to a higher percentage on the lower floors is counter to any idea of increased radiation for exposed upper floors and is attributable only to the direct effect of the chimney action. In particular the slope on the "mild" and "normal" days is markedly visible.

For numerical comparison the following tabulation is made based on lumping the figures for four approximately equal horizontal zones:

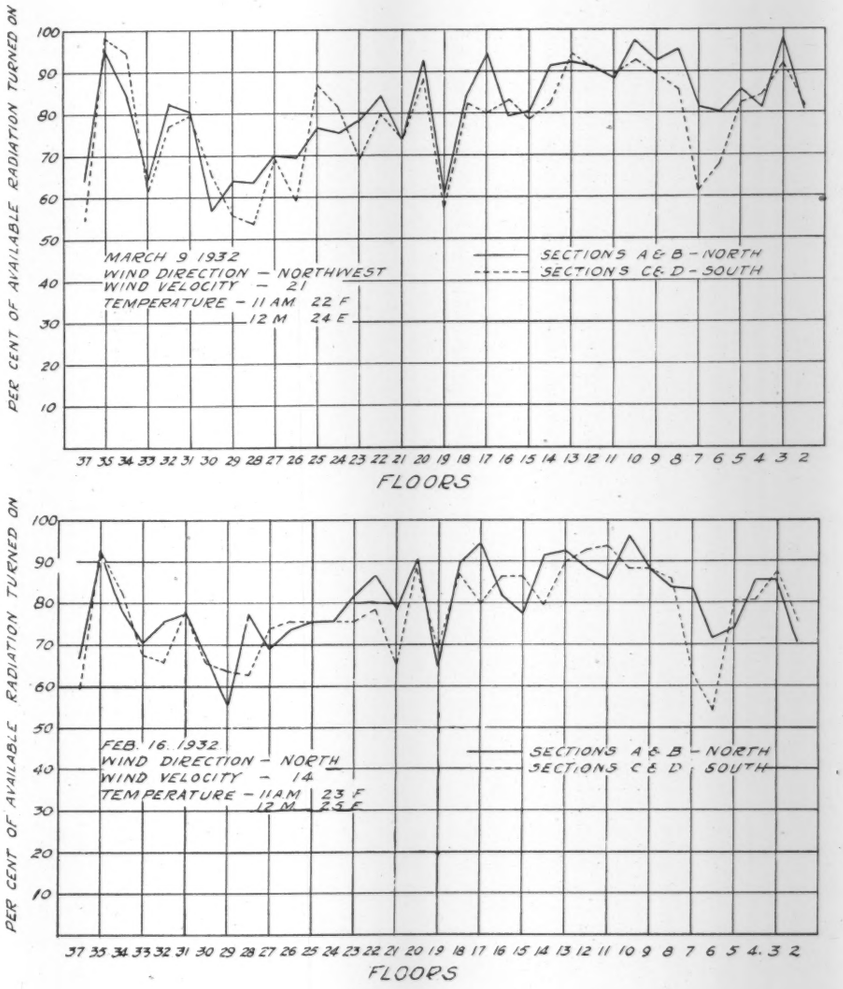
PER CENT OF TOTAL RADIATION IN USE

	Floors 2 to 10	*Floors 11 to 19*	*Floors 20 to 28*	*Floors 29 to 37*
Cold Days	83.1%	83.8%	75.7%	70.5%
Normal Days	71.6%	69.5%	54.8%	48.0%
Mild Days	46.0%	41.7%	30.8%	26.1%

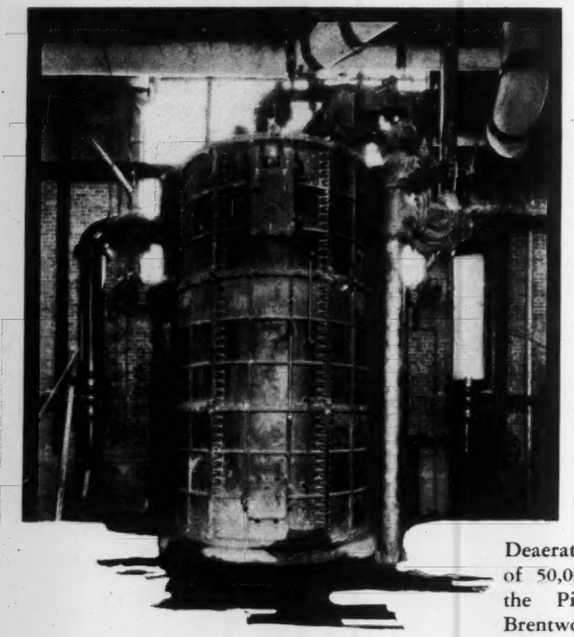
North Exposure Had No Effect

As an additional precaution in the design of a heating system, it is sometimes advised that the north exposure on the upper floors be given an increased radiation factor. In order to see what possible effect this north exposure had on the radiation actually put to use, the graphs in Figs. 8 and 9 were made. As will be noted these are plotted from the counts taken on the two cold days when the prevailing winds were from a northerly direction. Any extraordinary effect would be quickly noted under these circumstances, but the graphs do not indicate that the occupants on the north side felt impelled to use any more radiation than those on the south side.

These figures, graphs and observations are offered with the full realization that they are secured from one particular building and are, therefore, applicable (at least in their entirety) only to that building. Until further studies of more detailed nature are made on other buildings no specific rules can be stated.


PER CENT OF AVAILABLE RADIATION TURNED ON
MARCH 9 1932
WIND DIRECTION – NORTHWEST
WIND VELOCITY – 21
TEMPERATURE – 11 AM 22 F
12 M 24 F
SECTIONS A & B – NORTH
SECTIONS C & D – SOUTH
FLOORS
FEB. 16. 1932
WIND DIRECTION – NORTH
WIND VELOCITY – 14
TEMPERATURE – 11 A.M 23 F
12 M 23 F
SECTIONS A & B – NORTH
SECTIONS C & D – SOUTH
FLOORS

Figs. 8 and 9—Graphs made to find out effect of north exposure on required radiation. Occupants on the north of the building used no more radiation than those on the south.



Deaerating heater with capacity of 50,000 gallons per hour at the Pilgrim State Hospital, Brentwood, Long Island, which protects steel hot-water piping from corrosion. The water is used for domestic and general service requirements.

. . . . *By* C. E. Joos* and V. A. Rohlin†

Combating Corrosion of Piping

WITH the present stress on economy in the industrial and commercial fields, failure of piping due to corrosion and loss of capacity of piping from choking up because of the products of corrosion are receiving increased attention. Among the preventive measures which are available are the use of corrosion-resistant materials, chemical treatment of the water, removal of dissolved gases by heat, etc.

In an article published last month, the causes of corrosion were explained in some detail, and the meaning of *pH* value as an index of the corrosive tendencies of a water was outlined.

This paper deals with the treatment of the water supply by deaeration to prevent corrosion of water lines, such as condensate return lines, boiler feed and hot-water distributing and service piping and of economizers in boilers. As with most engineering problems, the correct answer is determined by the circumstances of the individual plant.

How Deaeration Protects Piping

Deaerating equipment affords protection to piping by eliminating dissolved oxygen and carbon dioxide. The removal of the carbon dioxide raises the *pH* value, but where low *pH* value is due to a mineral acid, a deaerator cannot be expected to correct the corrosive condition entirely. In infrequent cases, where mineral acids are present, rapid corrosion of cold water piping will also take place. Chemical correction of such water supplies may be made at the source to protect the water mains.

The deaerator is essentially a chamber in which water in finely divided form is brought into contact with steam by spraying or by flowing over trays. The water is heated to the boiling point by this intimate contact and the gases, being insoluble in water at this temperature, are removed from the apparatus by liberal venting. To accomplish complete removal of the oxygen and CO_2, the water is heated to the full temperature of the steam, bringing about a water vapor pressure equal to the total pressure of the atmosphere in contact with the water surface. If the water is desired at a temperature below atmospheric boiling point, 212 F, vacuum-producing equipment is employed for venting the steam-air mixture, or heat exchange apparatus is used. Thus if for hot-water service it is desired to maintain a temperature of 140 F, it is necessary to produce a vacuum within the deaerating heater corresponding to the boiling point at 140 F, or 24 in. of mercury.

To produce temperatures higher than 212 F it is only necessary to increase the steam pressure. Thus a deaerating heater operating at 60 lb. per sq. in. gage will deliver water having a temperature of 307 F. Deaerators for the delivery of water at such temperatures are used in the preparation of boiler feed in steam power plants or for special processes.

Types of Deaerators

Deaerators may be divided into three general classes, as follows:

Direct contact (tray or perforated reboiler)
Evaporator or surface type
Flash type

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In the **direct-contact deaerator** the water is heated and deaerated, as the name implies, in direct contact with the heating steam. For this reason, such deaerators are less expensive than is equipment which employs surface heating, but are not suited to all cases where the deaerated water is to be used for drinking or cooking,—that is, where the steam might contain injurious substances or gases, which might impart odor or taste to the water. They are very generally used for boiler feed and process work, and for domestic hot-water supplies where clean steam is available. Water enters the equipment through sealing and distributing boxes designed to prevent back flow of steam into the water lines, and to distribute the water over a tray stack. In flowing over the trays it is heated to the temperature corresponding to the pressure of the steam in the equipment. When this temperature has been reached, the solubility of dissolved gases is reduced to zero.

When the gases are driven out of solution they are still mingled in the form of small bubbles diffused through the water. Their release does not occur instantaneously and some means are provided for removing them. In some equipments the trays which serve for heating are followed by "air separating trays." In other equipments the final scrubbing action is performed by steam admitted through a perforated "reboiler" submerged in the water below the heating tray stack. Another design employs two shells, the heating tray stack and the air-separating tray stack being in separate shells. After heating in the upper compartment the water gravitates through a connecting pipe to the deaerator shell; steam under somewhat higher pressure is admitted to the downtake pipe, the purpose being to heat the water in transit so that it will be at the boiling point upon admission to the deaerator.

Such deaerators are usually equipped with float-controlled water inlet regulating valves and with automatic overflow equipment to prevent flooding, as from the admission of some uncontrolled supply. If the installation is to deliver water at a temperature lower than that corresponding to the pressure of the steam supplied, a steam control valve, pressure- or temperature-actuated, is included.

A small surface heater or vent condenser is utilized to reclaim the heat of and the condensate from the vent vapor, returning the heat to the water. Vacuum pumps or steam jet ejectors are employed for removal of vent gases from equipments operated at pressures below atmosphere.

In the **evaporator or surface deaerator** the steam for deaeration is supplied by evaporating some of the water by means of a steam coil or tube bank located below the heating or deaerating trays. In this way the steam supply, which may be exhaust steam carrying oil or grease, does not come in contact with the water, a point to be considered where the hot deaerated water is used for domestic purposes or for sterilizing, etc. Ordinarily, the water is first passed through a surface-type or closed heater external to the deaerator but supplied with steam from the same source as the evaporating element within the shell. Due to the low temperature of the water while traversing the preheater, the transmission of the heat is rapid. The water is preheated in the closed heater to almost its final temperature before it is admitted to the deaerator. In passing over the trays in the latter it is further heated by direct contact with the steam that has been generated by the evaporating element. The pressure of the steam supply must, of course, be somewhat higher than that corresponding to the final temperature desired, as temperature head is required to transmit the heat.

With this type of equipment no oil separator is used in the steam supply line to protect the water being heated from contamination, as the steam and water do not come in direct contact. A float-operated water control valve may be employed, and to prevent over-pressure a pressure relief valve is used.

A vent condenser is employed with a vacuum pump or steam-jet ejector if the apparatus is to be operated below atmospheric pressure. If the temperature of the steam supply is much higher than that at which the water is to be delivered a control valve, actuated by either temperature or pressure, is inserted in the steam supply line.

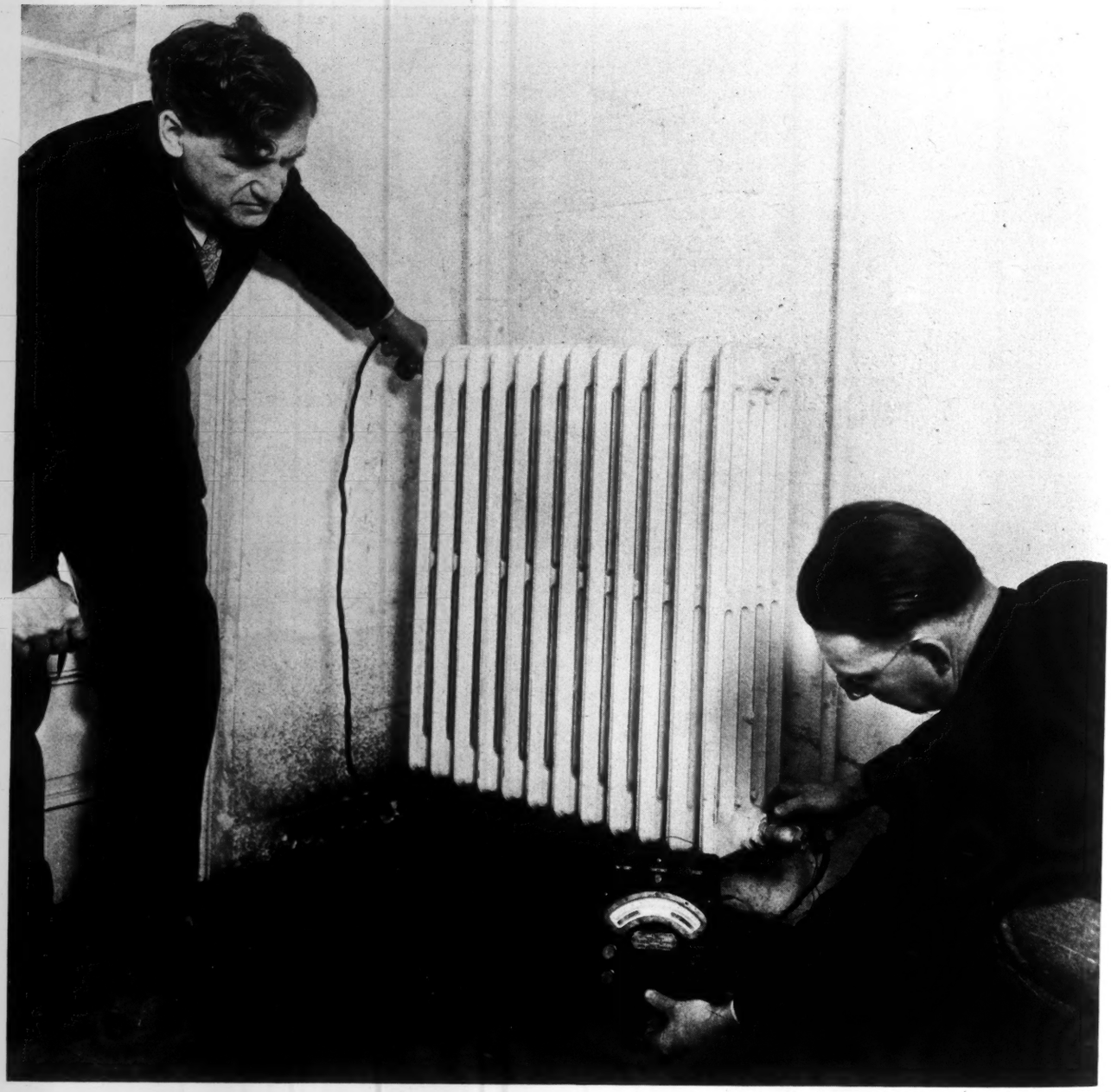
The **flash deaerator** is so called because the water is first heated to a higher temperature than that at which it is to be delivered, and then flashed or boiled by spraying it into a chamber under lower pressure. The heater may be of the direct-contact open type or the surface closed type. In either case the water is ordinarily heated to about 20-30 F above its final temperature and when admitted to the flash chamber boils instantly under the lower pressure. Thus the solubility of the gases is reduced to zero and the gases take the form of bubbles, which are removed by passing the water over a tray stack, as in the direct-contact deaerator. Equipments of this type include a vent condenser, preheater (either open or closed), and flash chamber, with water, steam and overflow controls.

Note: Methods of installing deaerators, typical examples, deaerating condensate returns, and deaeration in connection with central heating plants and for special industrial applications are covered in the next installment of this article.

● *"A Stitch in Time Saves Nine"*

This article tells how deaeration is used to eliminate dissolved oxygen and carbon dioxide in order to reduce the corrosive tendencies of water, and describes types of equipment and their operation.

Costly and troublesome replacements of piping, and loss of capacity due to choking up from the products of corrosion, can be reduced if proper attention is paid this important subject. It's another case of a stitch in time saving nine. ●



Left, the author. Right, Howard A. Cook.

Saves Fuel by Equalizing Flow in Hot-Water Heating Systems

Example Shows Annual Loss of $1.35 per Radiator in Unbalanced Job—Check-up Will Cut Cost

When some radiators in a forced hot-water heating system are too hot and some are too cold, fuel is wasted. In balancing such systems, the author uses thermocouples for finding the temperature drop through each radiator, as shown here. On jobs where radiators overheat a check-up before the coming heating season will save money.

• • • *By* Erwin L. Weber*

IN a properly designed and equalized hot-water heating system, each unit of radiation should receive its proportional share of the flow of water. This ideal condition will naturally produce the same drop in temperature in the water as it passes through the various units and will make the difference in temperature between the flow inlet and return outlet the same.

Forced Circulation Systems Do Not Tend to Equalize

In a gravity system each unit of radiation acts as a pump which produces a thermal motive force from the drop in the temperature of the water passing through the radiator. In a radiator with sluggish circulation the temperature drop is high and it therefore establishes an increased thermal motive force. Hence, there exists a natural tendency toward the equalization of flow. In a forced-circulation system such a tendency toward self-equalization *does not exist,* except as induced by the small part of the motive force which is present due to gravity action. The equalization in such a system should therefore be accurately provided.

Economics of Equalization

The economic advantage of correct equalization becomes apparent when the cost of lack of equalization is analyzed.

Consider a heating system in which the average drop in temperature through the radiators under a given weather condition is, or should be, from 175 F to 165 F, or 10 F. The average temperature is 170 F. Then, with a room temperature of 70 F the heat output will be 0.617 × 240 or 148 Btus per standard sq. ft. of radiation.

If now, as is often found in existing systems, a number of the radiators in the system show a drop of 25 F in place of 10 F, the rooms in which such radiators are located will be cold. To make these rooms comfortable, the average temperature of their radiators should be 170 F, which will necessitate raising the flow temperature to 182½ F.

The balance, or majority, of the radiators will then have an average temperature of about 177 F and windows will be opened in the overheated rooms. The heat output of these radiators will be 0.673 × 240 or 162 Btus per standard sq. ft. of radiation. (See p. 98, *A. S. H. V. E. Guide 1932.*)

Instead of supplying 148 Btus per sq. ft. of radiation, it is necessary to supply 162 Btus, or 9 per cent more heat than would be required if the flow of water through the radiators were properly balanced.

In an average system, under such conditions of unbalance, where the average radiator is of 50-sq. ft. capacity and the annual cost of fuel is 30c per sq. ft. the total cost of fuel per unit of radiation is $15.00 per year and the loss due to poor equalization is 9 per cent of $15.00 or $1.35 yearly. During an assumed 25-year life of the building and heating system, the cost of poor equalization becomes nearly $34.00 per radiator. This indicates that correct equalization is essential to economy of operation.

Thermocouples Determine Drop Through Radiator

Various devices are used for the purpose of equalization, but means of accurately testing such equalization are usually not employed. Frequently, the contractor attempts to determine the difference in temperature between the flow and return ends of the radiators by feeling them with his hands. Such attempts must be carried on at temperatures below 140 F, as above 140 the sensation of heat becomes a sensation of pain and differentiation is impossible. Hence equalization at working temperatures of 160 to 190 F cannot be secured by this method. Furthermore, such a method is bound to be far from accurate, as at one radiator the right hand feels the higher and the left hand feels the lower temperature, and at the next radiator, perhaps in another room, the conditions are reversed. Moreover, different degrees of dampness of the hands affect the accurate feeling of the sensation of heat. At best, such methods can give but crude results.

The writer has recently employed thermocouples to determine the drop in temperature through radiators by contacting one thermocouple at the inlet and one at the outlet of the radiators. The readings on the galvanometer then indicated the difference in temperature at these two points, equivalent to the drop through the radiator. By this means a very accurate equalization is obtainable. The process is similar to the equalization or adjustment of a ventilating system by the use of an anemometer as an indicator of the air movement.

Previous to equalization, the temperature drops through the radiators would vary from 4 to 30 F on the average system as tested.

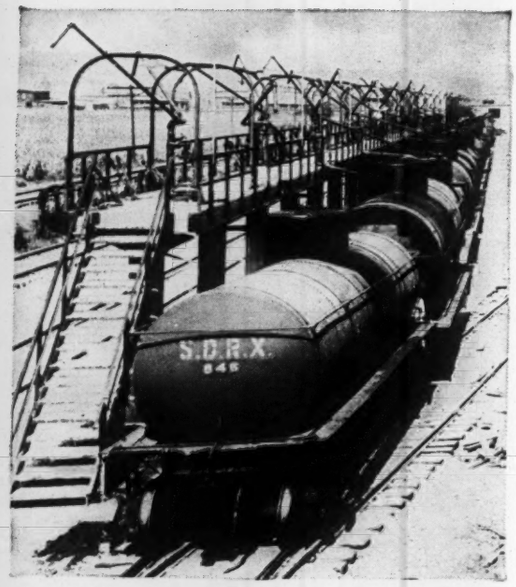
A satisfactory and economical method of equalization is with the use of valves or other devices, which can be externally throttled and set while the plant is in operation, analogous to the use of dampers or lock registers in a ventilating system. By measuring the air delivery at each outlet with an anemometer, the outlets can all be set to adjustment and balance. Similarly, by measuring the temperature drop through each radiator with thermocouples, the equalizing devices can all be adjusted to equalization and balance.

Usually it is only necessary to take two—or at the most three—sets of readings and make a similar number of adjustments to secure an acceptable balance.

The cost of taking such readings and making the required adjustments is considerably less than one dollar for each radiator, and will easily pay for itself in less than one year's operation through the fuel saved.

* Mechanical Engineer, Seattle, Wash.

PIPING STEAM *for* Heating Oil Tanks



Oil-handling stations and many industrial plants require provisions for heating heavy oils in order that they may be handled quickly and efficiently. Mr. Lewis made a survey of several steam piping lay-outs for this purpose; in this article he describes their good — and bad — points and tells how to arrange the piping to accomplish the desired results effectively and economically.

By Samuel R. Lewis*

SOME oils, especially those of the heavy fuel type, must be warmed in cold weather in order to be handled quickly and efficiently. Change in the bulk of oil due to temperature is of magnitude and it is customary for the dealer to purchase oil at a temperature of 60 F. If he then delivers it by bulk at a much cooler temperature he will lose by the shrinkage far more money than he must expend in order to heat the oil.

Up-to-date oil-handling stations have provision for heating the oil in the tank cars so that it can be removed quickly, and for heating it in the stationary storage tanks so that it will flow rapidly to the delivery trucks and so that its shrinkage due to chilling will be overcome.

The oil from the stationary tanks must be warm enough so that it will not chill below an easy-flowing temperature in the truck-tank, lest time of the latter be wasted when transferring the oil to the tank of the ultimate consumer.

Problem Requires Care

The problem may appear to be a simple one, requiring little thought, and, in fact, many of the early installations of oil tank heating were made in this manner. Sometimes an old boiler was secured, equipped with a second-hand oil burner, and piped up to steel coils built into the tanks. In many cases several of these coils were connected to a single supply main or to a single return main. The tanks which contained the heating coils often were scattered around the yard and the supply and return mains frequently were not protected adequately against freezing or against physical damage.

It might often happen that a cold snap caught the custodian unawares, with the result that water lying in a steam coil in an empty tank was frozen, bursting a pipe at a point where no one could see it or suspect its existence.

Joints May Be Steam-tight, But Leak Oil

A pipe joint which will be tight with water or steam may leak oil. Any threaded pipe coil in the design of which expansion and contraction have not been compensated for properly may leak oil, even though no freezing of water has occurred. Adequate and closely-placed supports for pipe coils inside tanks are important, as the coils may otherwise have water pockets which do not drain freely and which may cause much trouble.

Oil Must Not Enter Boiler

In consequence there seems to be a sincere lack of confidence among oil men in steam coils installed inside oil storage tanks. They discovered that excessive costs for fuel attended the operation of their tank-heating boilers. They found water in their stored oil tanks and oil in their boilers. A film of oil above the water in a steam boiler may account for greatly increased fuel consumption due to increased surface tension of the liquid above the water. When

*Consulting engineer, Chicago, Ill.

one tries to make steam by boiling oil, there is not much satisfaction with the heating system. If we were dealing with animal oil instead of mineral oil the boiler containing water and this oil would explode disastrously.

One large oil dealer, having experienced trouble, insists that the heating coils in his tanks shall not be connected back to the boiler. The boiler must take fresh city water, evaporate it into steam, send this steam to the tank coils, and the resulting condensed hot water with its possible increment of oil due to leaks, must, he insists, be wasted to the sewer.

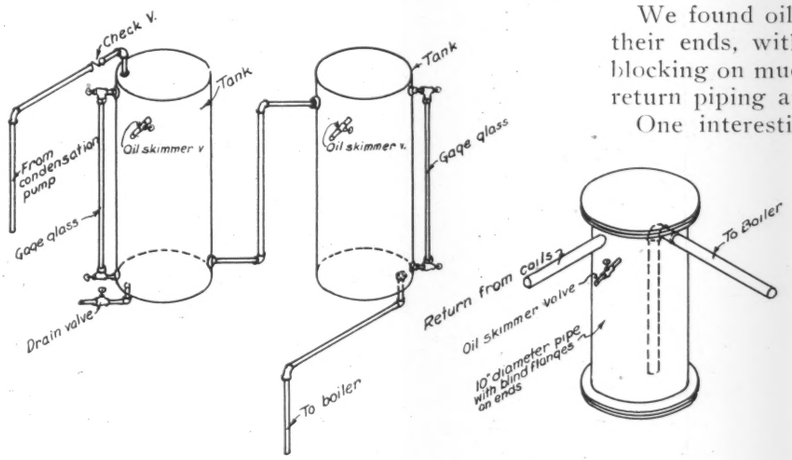


Fig. 1—Oil separator arrangement.

The two small vertical tanks at the left form a manually-operated scheme to prevent oil from leaky oil storage tank heaters from entering the boiler. An occasional opening of the oil skimmer valve (there are two stages to this separator) will remove much of the oil (it is hoped). The tank at the right is a single stage adaptation of the same principle.

Steam Should Be Controlled

No oil station customer would purchase fuel oil for heating without having provided a properly-insulated building which is to be heated and without equipment for preventing overheating following the combustion of too much fuel.

No oil station owner whom we visited, however, pays much attention to giving his enormous oil storage tanks any protection against too great cold, and we found no thermostats and very few thermometers in these tanks. Excess oil temperatures and unnecessary burning of fuel may be prevented at the fuel oil stations, but they are not prevented automatically, accurately, or definitely.

Arrangement for Skimming Oil from Water

We found several installations in which elaborate arrangements have been made to remove from the boiler water any oil which leaks into the steam piping. One of these schemes, using vertical steel tanks so that the lighter oil may be skimmed off above the heavier water by occasional manual valve-manipulation, is shown in Fig. 1.

Investigation Shows Variety of Methods and Lay-outs

We found oil storage tanks on their sides and on their ends, with solid foundations and with shaky blocking on mud. We found well-designed steam and return piping and we found poorly-designed piping.

One interesting installation is shown in Fig. 2. There has as yet been no trouble with the leaking of oil with this scheme; all joints inside the tank are welded, and there is opportunity to control the amount of heat by using the various valves to subdivide the heated surface. Since each separate coil has a separate trap the control is simplified. For such coils as this, extra heavy pipe is customarily used for the coils.

In one vertical storage tank installation the heating surface is built of threaded pipe in the bottom of the tank, with a flat steel baffle plate above. This baffle fits loosely, and causes the oil below it to achieve a rapid local circulation; this permits a much lower temperature in the main part of the tank. The draw-off connection is below the baffle where the warmer oil will be taken first, while there is ample area around the edges of the baffle for the heavier, cooler oil to pass down to the heater. The manhole in the tank is located so as to permit access both above and below the baffle. It was hoped that sludge would tend to collect on the baffle, from which it

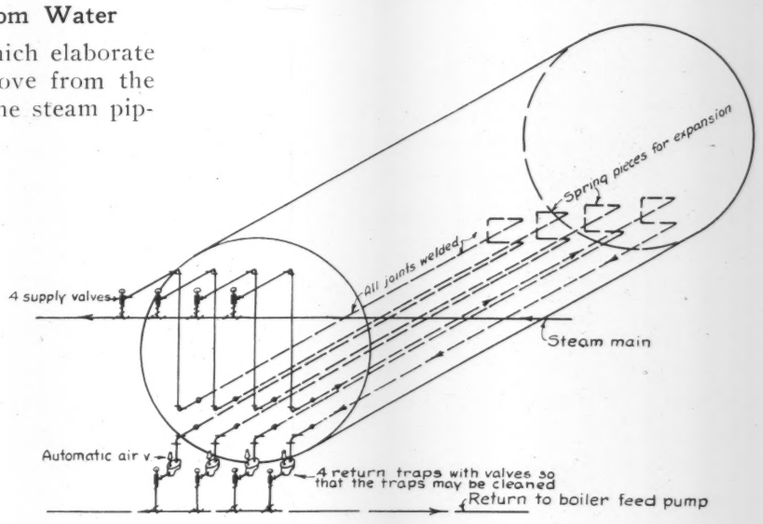

Spring pieces for expansion
All joints welded
4 supply valves
Steam main
Automatic air v.
4 return traps with valves so that the traps may be cleaned
Return to boiler feed pump

Fig. 2—A satisfactory steam heating equipment for a large oil storage tank.

The interior pipes are all welded. They drain freely all the way from the inlet back to the outlet as indicated by arrows. Provision is made for spring pieces to absorb the expansion and contraction, which may cause a difference in length of the two parallel pipes of as much as two inches in one hundred feet.

About 1½ sq. ft. of steam heated surface should be provided for each 100 gallons capacity in the tank for an installation like this.

Fig. 3 (Right)—An oil storage tank having a baffle plate-sludge pan above the spiral welded jointless steam coil.

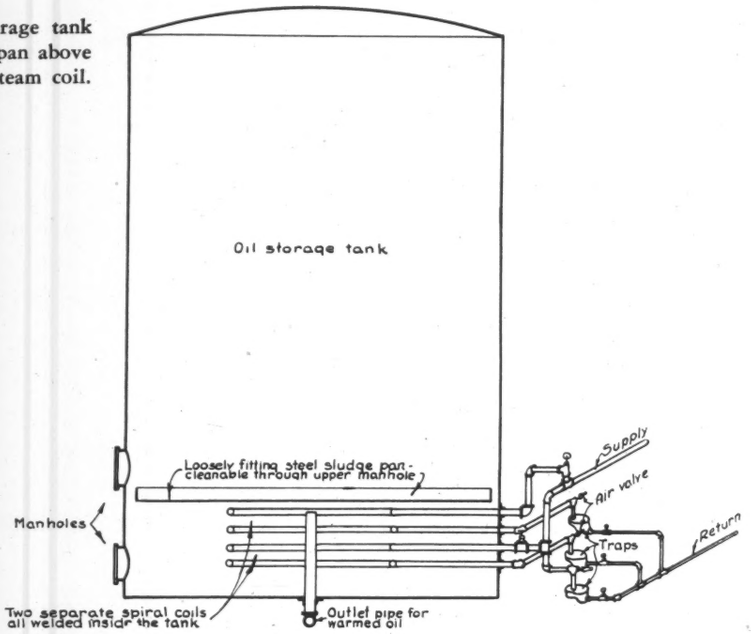
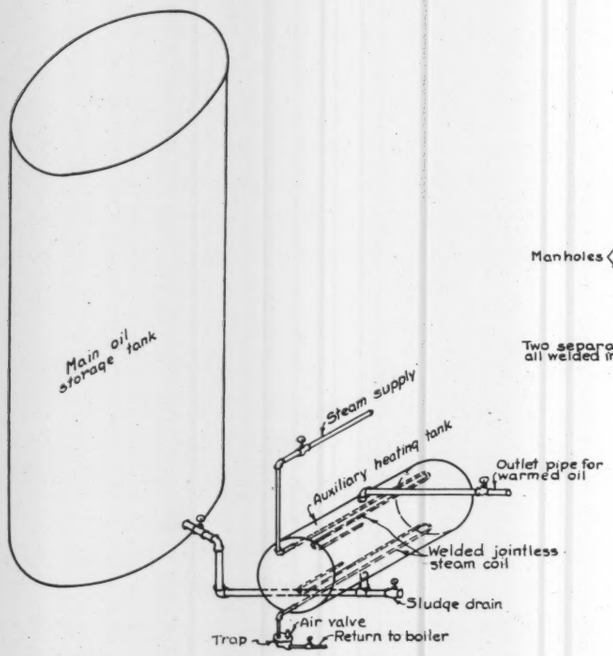


Fig. 4 (Below)—An excellent arrangement for an oil heater, using a comparatively small auxiliary tank which contains a jointless steam coil.



could be scraped off through the manhole door. Fig. 3 shows this scheme somewhat elaborated and improved.

We found one variation of this idea in which a smaller tank is placed alongside a large tank, connected therewith through a large easy-flowing pipe. The outgoing oil is withdrawn from the top of the smaller tank. The heating coil is placed in the smaller tank. Thus there is a lively circulation of oil in the smaller tank even when no oil is being removed but there is no circulation or heat loss from the larger one. The smaller tank easily can be shut off and drained for inspection of the steam coil. This scheme is shown in Fig. 4.

One large oil dealer houses his storage tanks within a warmed building and manages to get along fairly well without local coils inside the tanks. Such an arrangement is slow in heating up a large shipment of chilled oil, although such a shipment usually must be pre-warmed by steam coils in the tank car before it can be unloaded.

Another plan is to install steam pipes close against the tank bottom but entirely outside of it. There seems to be no real reason why this should not be a satisfactory recourse, if adequate insulation is provided to confine the heat against the tank and pre-

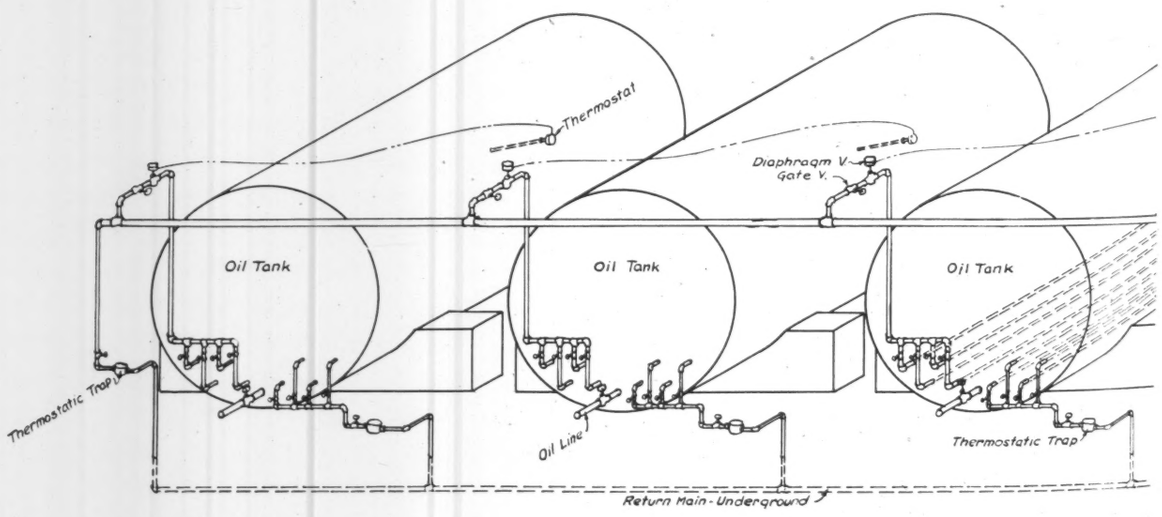


Fig. 5—A well-designed storage tank heating system.

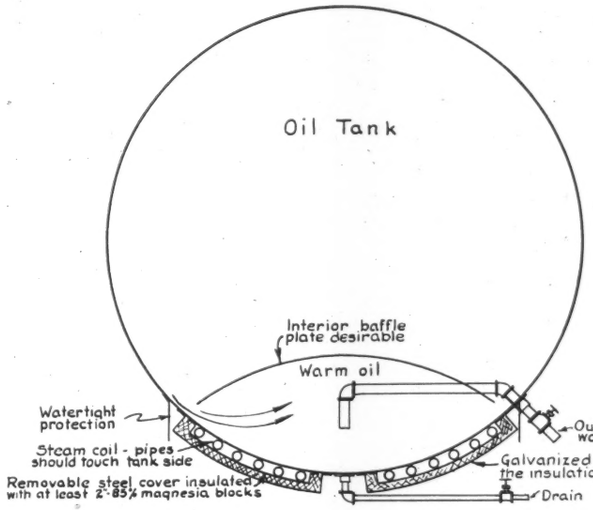

Oil Tank
Interior baffle plate desirable
Warm oil
Watertight protection
Outlet pipe for warm oil
Steam coil - pipes should touch tank side
Galvanized sheet steel cover to protect the insulation and to keep it assuredly dry
Removable steel cover insulated with at least 2"-85% magnesia blocks
Drain

Fig. 6—An exterior steam tank heater.

The oil may be heated by steam pipes in contact with its outside. The pipes must be boxed in, and must of course allow for expansion and contraction and for perfect drainage and air venting. A loosely-fitting baffle to limit the circulation of the oil inside the tank and to concentrate the warm oil near the outlet is desirable. If the tank is out-of-doors all moisture must be kept off the insulation.

vent its waste. Fig. 6 shows such construction; the hot pipes should touch the tank sides.

One Good Arrangement

In Fig. 5 is shown an oil tank heating system with welded interior steam coils, and with the steam and return piping and valves and other auxiliaries. The steam piping may be run out-of-doors, may be overhead on posts or may better be carried underground in approved insulating tile or in a masonry conduit, and should be easily accessible for inspection and repair.

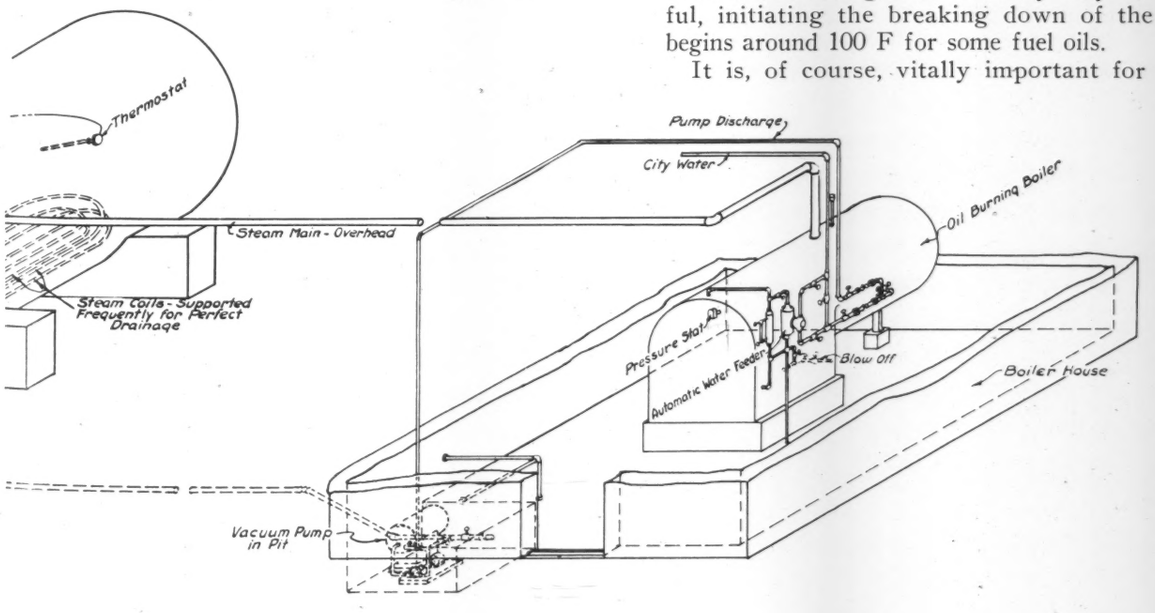
The design and installation of this piping must be done with care. There must be no accumulation of condensed water in the supply main, and it must be possible to drain the return main rapidly and positively. Both mains must be thoroughly protected against getting wet. It is comparatively easy to double-insulate and to water-proof a straight length of piping out-of-doors, but it is difficult to protect the raw ends of the insulation at valves, traps, etc. Therefore it is a good idea to have the heads of the heated tanks project into a heated building, or to build shanties at the heated tank-ends to protect the valves and traps against frost and unauthorized manipulation.

It is advisable to insert liquid thermostats in tanks at the most vital points, depending on the location of the outlet connections, for controlling the temperatures of the steam coils. If all of the steam coils are so protected, the oil burner may have a simple steam pressure control, or better yet, if the boiler serves only as a storage tank heater there is available a simple little electrical hook-up whereby the oil burner will operate until the last thermostat has closed off the supply valve to the heater in its particular tank, no matter which one this may be, and will open up again when any one of the tank thermostats resumes the open position.

It is probable that if all storage tanks were placed inside well-insulated buildings, economies against overheating in summer and under-heating in winter would be achieved.

Excess temperature such as may be due to bright sunshine does the oil no particular good. Excess temperature such as easily can be obtained by uncontrolled heating coils certainly may be very harmful, initiating the breaking down of the oil, which begins around 100 F for some fuel oils.

It is, of course, vitally important for the person



who installs a steam coil in an oil tank to have means of getting at that coil. If a steam coil in a large tank goes wrong when the tank is full of oil very expensive and annoying results may ensue. Every coil should have a valve on each end, so that it may be isolated to prevent oil waste and so that the other coils may continue in service.

Oil As Heat-carrying Medium

The use of heated oil pumped from a steam-heated tank in the boiler room to heating coils in outlying exposed storage tank heaters, has many advantages. With such an arrangement there is no danger of freezing. If a little leakage occurs from oil to oil no great harm will result, and such leakage would be manifest

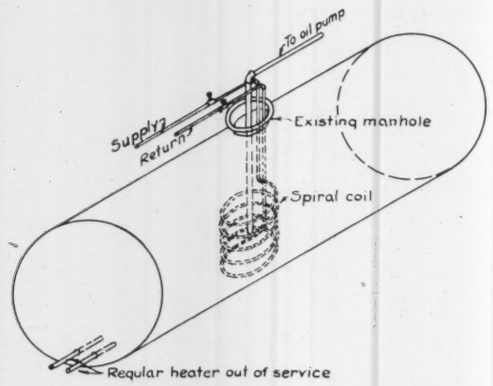


Fig. 7—Emergency heater to warm oil.

A spiral copper coil was threaded in through the manhole and was lowered to the bottom of the tank. A temporary outlet pipe was connected down to the center of the spiral coil so that it would catch the warmed oil. Steam under enough pressure to overcome the static head was forced through the coil, blowing the condensation up through the return. Hot water or hot oil under pressure could, of course, have been pumped through the coil.

at the expansion tank. If leakage occurs from oil to water or vice versa, in the heat transfer tank in the boiler room, this also would be noticeable very quickly and could be corrected quickly and economically, since the oil heating tank will be very small. Two of these transfer-heaters for alternate use would be desirable.

Steam Pressures Used

In the dozen or more plants which are the basis for the observations made in this paper, the operating steam pressures varied from that of the atmosphere to 125 lb. When low pressure was used the boilers were full of oil. When high pressure was used the oil tanks were full of water. We admit that both of these statements are a trifle salty, but the truth is that there was, in the observed oil storage plants, no reasonable explanation for the great steam pressure variation which was found.

It is not practicable to vary the steam pressure in each case to stop leakage by balancing exactly the varying oil pressures due to changes in the height of the oil in each tank. A comparatively low steam pressure is more economical of fuel than is a high steam pressure when the entire function of the steam is to heat the oil.

VOLUME OF "1,000 GAL. OF OIL AT 60 F" AT VARIOUS TEMPERATURES

F	GALLONS
30	988
40	992
50	996
60	1000
70	1004
80	1008
90	1012
100	1016

It would require approximately 200 *lb. of steam to raise* 1000 *gal. of oil* 50 *F in an hour.*

Since a pump usually is necessary with any type of heating system used for heating oil in storage tanks, advantage lies in using comparatively low steam pressure with thermostatic vacuum traps on the outlets from the coils and for draining the mains, with an electric vacuum pump placed low enough to receive all condensation by gravity.

With such a plant there is minimum opportunity for condensation remaining in mains or in heater coils, where of course it will promote water hammer. It is not unlikely that water hammer due to inefficient drainage, partly closed valves, poor supports, etc., largely is responsible for broken joints and oil leaks in existing oil tank heating systems. Well-designed vacuum return arrangements would eliminate this.

Radio announcers and engineers covering the recent national political conventions were kept comfortable in their cell-like, sound-proof broadcasting booths high on the mezzanine floor of the Chicago Stadium by unit air conditioners.

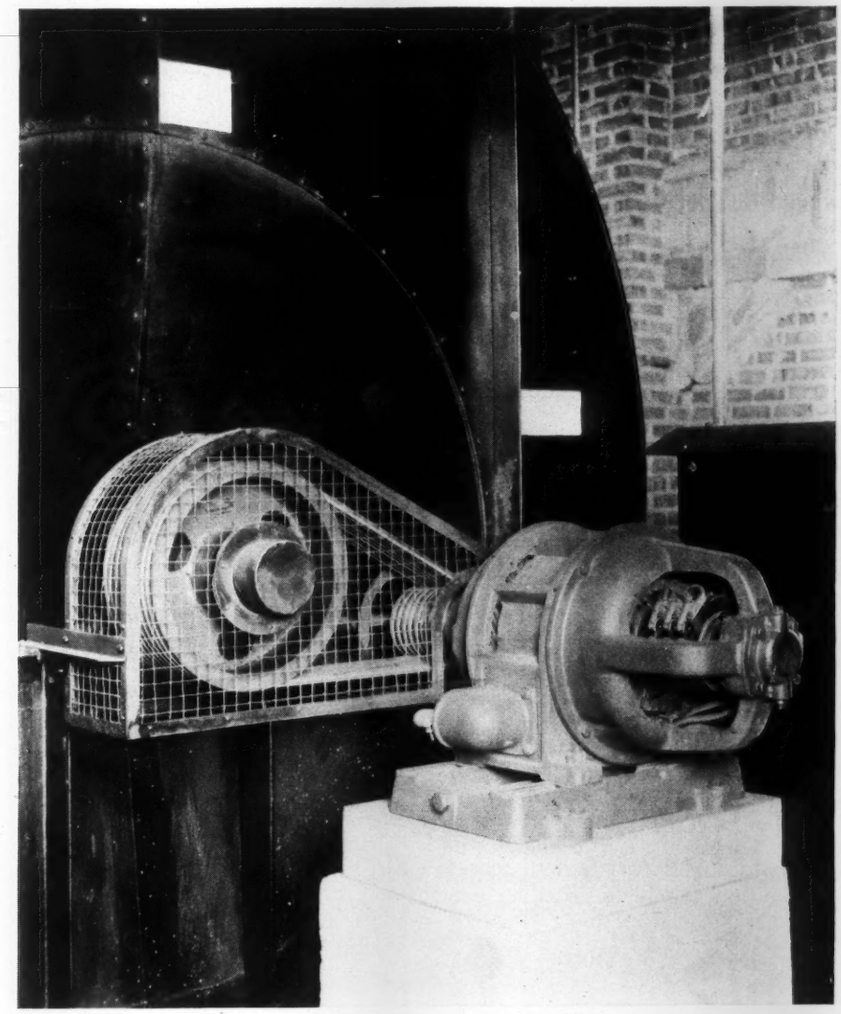


Fig. 1—V-belt drive for transmitting 20 hp. from a motor running at 1750 r. p. m. to a ventilating fan running at 560 r. p. m. The method of selecting this drive is shown here.

Choosing a Multi-V Drive for a Fan

Success of a drive installation depends upon a conservative consideration of the engineering factors which enter into drive selection. These factors are outlined in this article, and the steps in selecting a V-belt drive for a ventilating fan shown.

MULTIPLE-V belt drives are used extensively with heating, piping and air conditioning equipment such as fans, blowers, pumps, compressors, etc. Proper selection of such a drive should be based on a knowledge of the factors which are involved in order to insure its having a maximum life and economy in maintenance and operation.

The following information should be available when a multiple-V drive is selected:

The horsepower to be transmitted (based on the horsepower of the motor).

Speeds of the driving and driven units.[1]

[1] The speed ratio may be figured as equal to the ratio of the pulley diameters, although to be exact the speed ratio is equal to the ratio of the pitch diameters.

Center-to-center distance of the sheaves desired (with plus and minus distances).

Characteristics of load (constant, overload or starting peaks, frequency of starts, etc.).

Space limitations.

Shaft dimensions.

Manuals are available from manufacturers of V-belt drives which give complete engineering information enabling the selection of the proper drive for almost any application desired. Tables listing stock drive combinations for various motor speeds, horsepower and speed ratios allow for the factors outlined above and greatly simplify selection.

The number of belts to be associated with the belt and sheave combination is determined by dividing the manufacturer's rating per belt (for the belt velocity and arc of contact characteristics of the drive) into the desired horsepower capacity. Use of a sufficient number of belts of the proper cross-section is important so that each will operate at the tension for which it was designed, avoiding over-stressing the belt and possible slippage in the sheave grooves which might produce high temperatures to the detriment of the life of the belt and its efficiency. Five belt cross-sections have been standardized and are known as Nos. *0, 1, 2, 3, 4* (or *A, B, C, D, E*) with these dimensions, respectively: ½ in. wide by 11/32 in. deep, 21/32 by 7/16, ⅞ by ⅝, 1¼ by ¾ and 1½ by 1.

The characteristics of the load, such as starting torque and overloads, may increase the work which must be done by the drive, and this is allowed for in the original selection. For a fan, for instance, the horsepower to be transmitted should be increased perhaps 25 per cent in choosing the drive. Load characteristics of other types of equipment may increase the required capacity of the drive by 100 per cent.

The ratio of the sheave diameters is determined by the ratio of the speeds. When a belt is bent around a sheave, its outside layer is under tension and its inner layer is in compression; in the manufacture of V-belts this is, of course, taken into consideration. The smaller the sheaves of the drive, the greater will be the stresses set up in the belt and minimum sheave diameters for the different sizes of belts are listed by manufacturers.

The arc of contact influences the horsepower which a belt can transmit. A correction factor may be applied to the horsepower rating, being 1.00 for a 180-deg. arc of contact and ranging down to approximately 0.85 for an arc of 120 degrees. Arc of contact is figured with the formula:

$$\text{Arc of contact} = 180° - \frac{(\text{Diam. large sheave} - \text{diam. small sheave}) \times 57}{\text{Center distance}}$$

To point out the method of using the manuals which are available a drive for a ventilating fan is selected in the following example, one such drive manual being referred to; the drive is illustrated in Fig. 1:

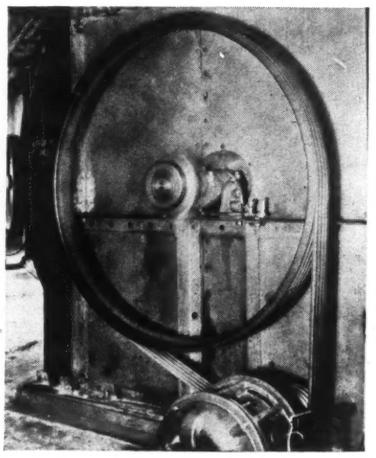
It is desired to transmit 20 hp. from a 1750-r.p.m. motor to a blower running at 560 r.p.m. (The full-load motor speed is considered so that the desired speed of the driven unit under load will be obtained.) A center distance of approximately 25 in. is specified.

Shaft dimensions and space limitations do not enter into the original engineering problem. Shaft dimensions, of course, must be known so that sheaves with suitable bores and key-ways will be furnished. Space limitations must be known so that the sheave diameters selected for the drive are within them. As a general rule, standard drives for ventilating equipment conditions are well within the limits placed on the drive by the surrounding space.

From the manual it is learned that for this speed and horsepower condition a drive with 21/32 in. x 7/16 in. cross-section belts will be most practical. We will figure on a drive with a capacity of 25 hp., which is 125 per cent of the 20 hp. to be transmitted. The ratio of sheave diameters as determined from the ratio of the speed is 3.13. Referring this ratio with the approximate center-to-center distance to that part of the manual showing belt and sheave combinations for the belts selected, we obtain the combination of a 6.4-in. pitch diameter driving sheave, a 20-in. pitch diameter driven sheave, and belts with a pitch length of 91.4 in., which will effect a center distance of 24 in. The engineering data for the belt and sheave combination give a horsepower rating per belt of 2.88 where this speed-reducing combination is operated from a 1750-r.p.m. motor. Dividing this rating per belt into the 25 hp. drive capacity as desired indicates the use of 8.7 belts, or nine belts. Thus we conclude our selection, which, of course, will be checked with the space limitations to assure that it fits the job.

The V-drive manufacturer will furnish a print showing the detail dimensions of the drive which is being furnished. On this print, an adjustment will be indicated for the center distance, so that it can be decreased in order to assemble the belts onto the sheaves and increased in order to compensate a slight amount of belt stretch characteristic of V belts. The maximum stretch in a V belt is usually 5 per cent, so that the maximum adjustment that will be necessary on the center distance is about 5 per cent, 1 per cent of which will be a decreasing adjustment to assemble the belts, usually provided by means of a sliding base under the motor.

While the selection of a multiple-V drive can in most cases be easily made by reference to tables as outlined here, the nature of some loads requires special consideration. A flywheel on the driven load will usually relieve the belts of shock loads, while if this flywheel effect is in the motor the load on the drive will tend to be increased.



ON THE JOB

. . . a department devoted to short, practical articles on the installation, operation and maintenance of air conditioning, heating and piping equipment . . . . . .

Cutting the Cost of Extras

. . . . . By E. C. Parker*

ON A recent job eighty per cent completed, the owners decided to incorporate in the project a section of one floor devoted to a photographic bureau. In this bureau were to be located ten photostat machines, a dark room, chemical mixing sinks, etc. Located on several additional floors below were eight rectigraphs. For all these machines it was necessary to provide fresh water at 70 F with an allowable fluctuation of 2 degrees plus or minus, for use in an automatically-timed developing and washing process.

The only cold water service near the installation came from a house tank several floors above which was filled by the condenser discharge from the refrigerating equipment located in the basement and, during a three-month period when the peak load would be on the refrigerating equipment for the air-conditioning system, the water would reach a temperature of 83 F. Obviously, this supply could not be used.

Several propositions were investigated but could not be used because of the expense involved.

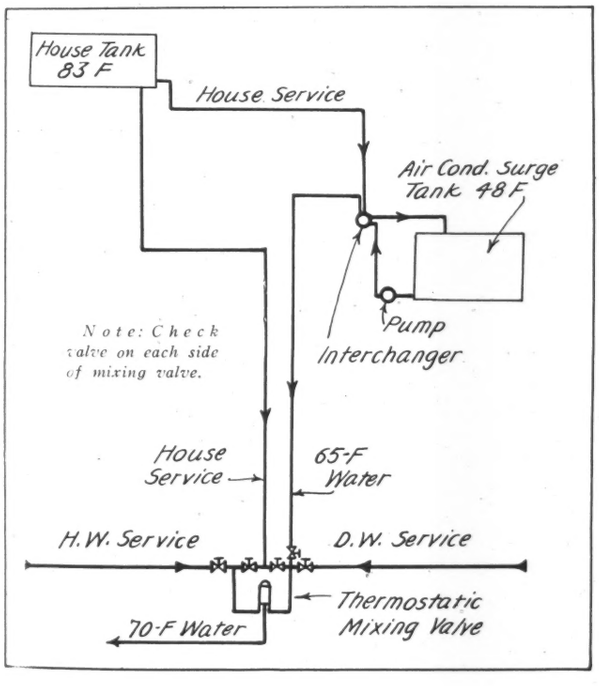
As the air-conditioning system included in its design a safety factor that would allow an added 24-ton refrigerating load, the following scheme was used:

A supply line from the house tank with 83-F water was run down to an interchanger, located just above the air-conditioning surge tank. The cooling was accomplished by installing a 2-hp. single-stage pump, with remote control located in the photo bureau, to pump 75 g.p.m. of this return water in the surge tank (at a temperature of 48 F) up to and through the interchanger, returning through the top of the surge tank. From the cooling coils in the interchanger, a supply line was run to a thermostatic mixing valve located close to the bureau.

As the photo bureau is forced to work at night and over week-ends when the air-conditioning system is shut down, a tap was taken from the drinking water supply to supply 20 g.p.m. at 45 F during such times. During the winter when the air-conditioning refrigeration is not in use the water in the house tank will be below 70 F, so a hot-water line was run to the mixing valve to meet this condition.

A line from the house tank with a by-pass around the mixing valve, as shown in the diagram, completely solved the problem of supplying 70-F water at all times. The by-pass valves need be set only for season or emergency and thereafter the operation is automatic. This method cut the cost of other proposed arrangements fifty per cent.

*Starret Bros. & Eken, New York City.



To furnish 70-F water for a photo developing and washing process this piping scheme was worked out, utilizing part of the air-conditioning plant's capacity to cool the water when necessary. Half the cost of other possible arrangements was saved this way.

Apply Pipe Wrench Properly

. . . . . By John J. Harman*

IN SPITE of the extensive use of the pipe wrench little attention has been paid to the proper method of applying this tool to the pipe in order to secure the best results.

It is self-evident that in order to insure taking a secure bite on the pipe wall that will not slip, the wrench must exert a certain amount of crushing force; if this force is excessive the pipe may be permanently injured. Flattening and indentation of pipe due to excessive crushing action of the wrench, particularly in pipe sizes 2 in. and smaller, is not an uncommon occurrence and yet few have realized how vitally the crushing action of the wrench is affected by variations in the angular position of the jaw and the location of the pipe in the gripping surfaces.

Direction and Magnitude of Forces

The force diagram shown here illustrates the forces that are brought into play when a 14-in. wrench is applied to 1-in. pipe in various position with a pull of 100 lb. on the handle at a distance of 10⅝ in. from the center of the pipe.

The diagrams superimposed on the outline of the wrench show the lines of action of the various forces and the upper diagrams show the magnitude of the forces. The AB force is the 100-lb. turning force, while AC_1, AC_2, AC_3, and AC_4 represent the forces exerted on the pipe by the wrench jaw and BC_1, BC_2, BC_3, and BC_4 represent the forces exerted on the pipe by the wrench bar.

The maximum crushing force on the pipe is determined by resolving BC_1, BC_2, BC_3, and BC_4, which are always somewhat greater than AC_1, AC_2, AC_3, and AC_4, into their tangential turning components C_1D_1, C_2D_2, C_3D_3, and C_4D_4 and their radial crushing components BD_1, BD_2, BD_3, and BD_4.

The diagrams indicate that the **crushing force is minimum in position No. 1,** when the pipe is gripped in a forward position and the jaw is tilted forward, while in position No. 4, when the pipe is back against the jaw and the jaw is in median position, the crushing force is increased to its maximum value.

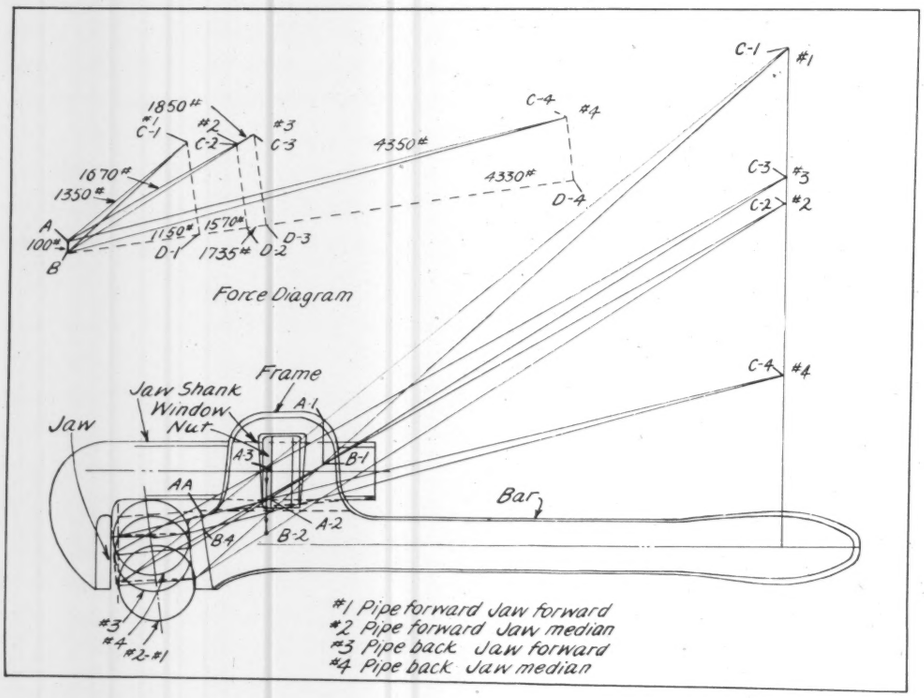
Due to the fact that the teeth bite into the pipe and the pipe wall yields to a certain extent under the pressure of the wrench, the jaw deflects forward when the load is applied and consequently the median positions of the jaw in the figure really represent extreme backward positions when the wrench first takes its bite before application of the load, and likewise the forward positions of the jaw represent median initial positions.

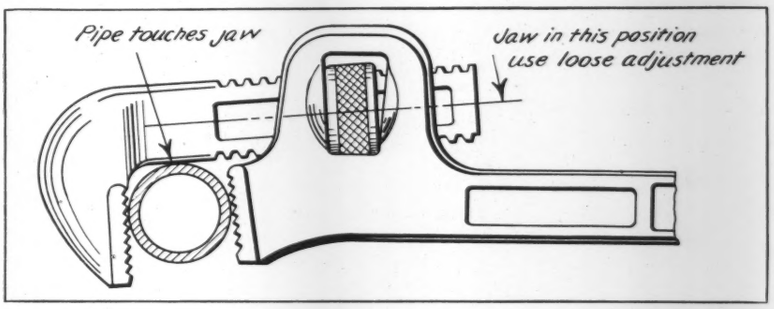
High Crushing Position Should Be Avoided

The great difference between the maximum crushing force of 4330 lb. and the minimum crushing force

*Assistant chief engineer, Walworth Company, when this article was written.

Force diagram for a 14-in. pipe wrench applied to a 1-in. pipe. Crushing force is a minimum in position No. 1 while in position No. 4 it is at its maximum value.





Proper method of adjusting pipe wrench to alloy cast-iron pipe to avoid excessive crushing force.

of 1150 lb. indicates the desirability of avoiding the maximum crushing positions, particularly for thin-walled pipe that may be seriously indented and for pipe of low ductility that may be cracked by extreme crushing action of the wrench.

Use of alloy cast-iron screwed-joint pipe for corrosive service conditions has emphasized the need for control of the crushing action of the wrench, because instances have been reported where 1½-in. and 2-in. cast-iron pipes have been broken by pipe fitters in making up joints; in some cases incipient cracks have been started which were not discovered at the time but later resulted in a failure.

The standard threading tolerance on both pipe and fittings is plus or minus one turn from the gauging notch; with this in mind it is apparent that when a pipe threaded one turn large is screwed into a fitting threaded one turn small, the turning force required to screw all the threads into the fitting is very great. Cast-iron pipe is so rigid that it can be compressed but very little, and it is quite impossible to bury all the threads in the fitting when the pipe happens to be threaded oversize and the fitting undersize.

Application of Wrench to Alloy Cast-Iron Pipe

Aside from the fact that cast-iron pipe threads cannot be made up as far as steel-pipe threads, cast-iron pipe is prone to injury by crushing action of the wrench because of the tendency to flatten the circular cross-section and crack the pipe wall longitudinally. Extensive tests have shown that when the wrench is applied to cast-iron pipe with the jaw shank resting firmly on the pipe wall, the three-point contact secured in this way is a material aid in resisting the tendency to flatten the circular cross-section, and the danger of cracking the pipe is greatly reduced.

Consequently, in order to avoid injury to the small sizes of cast-iron pipe by excessive crushing action of the wrench, a very loose adjustment of the wrench should be used. The wrench should be placed on the pipe so that the jaw shank rests on the pipe as shown in the diagram above, but care should be taken not to deflect the wrench jaw backward to the high-crushing position No. 4 in the first diagram.

The wrench adjusting nut should then be tightened just enough to make the wrench bite. This will insure gripping the pipe in the moderate crushing position No. 3, in addition to giving the pipe wall support by the three-point contact. If a close adjustment of the wrench is used so that the jaw cannot tip forward, the crushing action will be greatly increased and the pipe may be broken.

Do Not Use Oversize Wrenches

Another point that needs attention is the proper size of wrench to use for the different sizes of pipe. It is often stated that each size of wrench is suitable for pipe up to the maximum size the wrench will span, although in some cases both the maximum and minimum pipe sizes are specified. A typical listing is as follows: 6-in. wrench—pipe sizes ⅛ in. to ½ in., 8-in. wrench—pipe sizes ⅛ in to ¾ in., 10-in. wrench—pipe sizes ⅛ in. to 1 in., 14-in. wrench—pipe sizes ¼ in. to 1½ in., 18-in. wrench—pipe sizes ¼ in. to 2 in., 24-inch wrench—pipe sizes ¼ in. to 2½ in., 36-in. wrench—pipe sizes ¼ in. to 4 in., and 48-in. wrench—pipe sizes 1 in. to 6 in.

Obviously when a large wrench is applied to a small pipe the relatively enormous turning moment that is made available by the long moment arm, may greatly increase the crushing force exerted on the pipe and for this reason the use of oversized wrenches should be avoided.

Schedule of Proper Wrench Sizes

In the writer's opinion a desirable schedule for pipe sizes would be: 6-in. wrench—pipe sizes ⅛ in. to ½ in., 8-in. wrench—pipe sizes ⅜ in. to ¾ in., 10-in. wrench—pipe sizes ½ in. to 1 in., 14-in. wrench—pipe sizes ¾ in. to 1½ in., 18-in. wrench—pipe sizes 1¼ in. to 2 in., 24-in. wrench—pipe sizes 1½ in. to 2½ in., 36-in. wrench—pipe sizes 2½ in. to 4 in., and 48-in. wrench—pipe sizes 3½ in. to 6 in.

The writer believes that the above schedule should be adopted for pipe of all kinds, and that pipe fitters should be instructed in the proper application of the wrench so as to avoid the high crushing positions. This would reduce the danger of injury to the pipe, fitting and joint threads by excessive forcing of the joint and the danger of injury to the pipe by improper application of the wrench.

M. Barry Watson has prepared a chart which makes it possible to determine quickly the size of pipe necessary to use as a beam to support tanks, heaters, etc. Watch for it in an early issue.

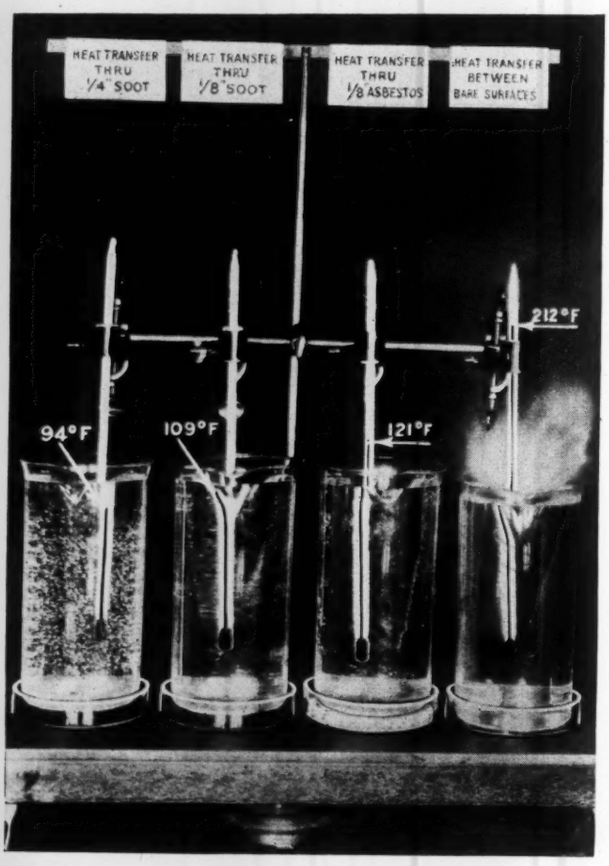
Clean Boilers Save Fuel

. . . . . *By* G. L. Davis*

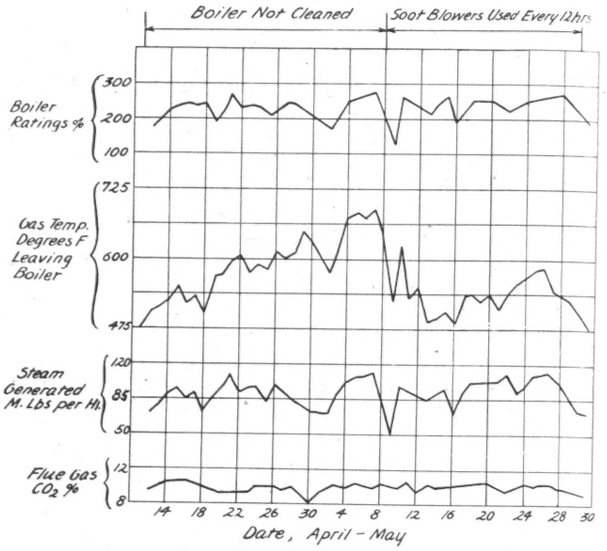
OPPORTUNITY for realizing heating-plant economy lies in maintaining clean boiler heating surfaces. Investigation has shown that soot's non-conductive properties are a fruitful cause of fuel waste; a perfectly clean boiler tube ⅛ in. thick will transmit approximately 40 times as much heat as will a tube lined or covered with a 1-in. thickness of soot.

"What price soot" is graphically illustrated in the photo reproduced here. The electric hot plate operating at 750 F has brought the water in the right-hand, uninsulated beaker to the boiling point, 212 F, while that in the other beakers, insulated by soot or asbestos, is in each case far below that temperature.

The chart shown covers the results of a test on a coal-fired boiler. On April 12, the boiler was put on the line after first thoroughly cleaning the tubes. From this date until May 8, 26 days, it was not cleaned. At the end of the period, flue-gas temperatures had risen to 690 F. It was then cleaned with mechanical soot blowers and the average temperature of the flue gases fell to 530 F. It remained at approximately this average throughout the remaining period of the test from May 8 to 30, during which time the soot blowers were used every 12 hr.



Experimental set-up showing how soot cuts down heat transfer and demonstrating the need for clean boiler surfaces if economy is to be achieved.



Results of test on a coal-fired boiler. Cleaning it by means of soot blowers reduced flue-gas temperatures to 530 F and gave a 5 1/3 to 8 per cent saving in fuel.

The fuel saved by reducing flue-gas temperatures amounts to about 1 per cent for every 20 to 30 F drop, depending upon the amount of excess air in the gases. It will thus be seen that thorough and systematic cleaning resulted in a 5 1/3 to 8 per cent fuel saving.

*Diamond Power Specialty Corporation, Detroit, Mich.

Text on Refrigeration Published

The third edition of "Principles of Refrigeration," by W. H. Motz, has recently been published by Nickerson & Collins Co., 433 N. Waller Ave., Chicago. Its 1032 pages are divided into 24 chapters and include 322 illustrations and over 100 tables.

The book has been adopted by the *National Association of Practical Refrigerating Engineers* for study in its national lecture course.

A 60-page chapter is devoted to the cooling and conditioning of air, and includes a discussion of the fundamentals, refrigeration requirements, methods, etc.

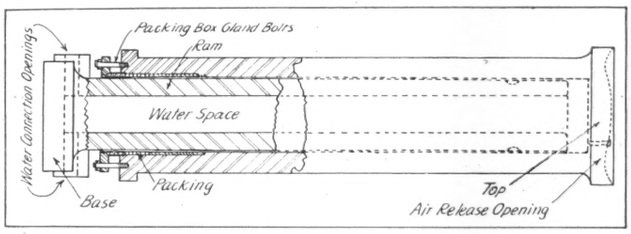
The other subjects treated are: elementary refrigeration systems; fundamental units and physical laws; refrigerating media; general principles of the compression and the absorption systems; the ammonia compression system; the absorption and other refrigeration systems; heat transmission in insulation and apparatus; methods of distribution of refrigeration; automatic systems; manufacture of ice; cooling, solids, liquids and gases; cold storage of commodities; erection, testing, operation and care of apparatus; working temperatures and pressures; prime movers; economics; and general considerations.

Don't Forget *Safety* in Piping!

. . . . *By* W. H. Wilson

SAFETY is one of the important factors in the successful and economical operation of industrial plants. Accidents, causing loss of property and injuries to workmen, interfere with the regular working schedule, increase costs and are expensive in other ways. Pipe work and pipe maintenance are largely in the scope of safety measures and careful working conditions.

Accidents may be caused by the expansive force of air and gases, as illustrated by the sketch of the heavy cast-steel accumulator (used in connection with high-pressure hydraulic power equipment) shown here. This accumulator is of the moving cylinder type, for installation in a vertical position.



Cast-steel accumulator of the moving-cylinder type. Air pressure in the water space forced the ram out suddenly; it would have been safer to use water pressure to remove the ram.

Previously, it had been in use at one location, and before erecting at a new location it was necessary to remove the ram for machining and other alterations. The accumulator, weighing several tons, was placed on supports, in a horizontal position. The packing box gland bolts were loosened, the water connection openings plugged and compressed air pressure at 100 lb. was put in the water space at the air release opening for the purpose of forcing out and removing the ram. When first applied the air pressure failed to start the ram and it was decided to give up this method. The air pressure —which was supplied by a hose connection—was shut off and suddenly, without warning, the ram shot out of the cylinder and landed on the ground. Fortunately none of the workmen were in front of it at the time. The air pressure stored in the water space of the ram was sufficient, after the friction of the old packing was broken, to eject the ram in a speedy manner. When the air supply was shut off the workmen had no control over the expansive power "stored" in the accumulator.

It would have been safer to use water pressure to force out the ram or to fill the accumulator with water and use a small size hose to admit the compressed air.

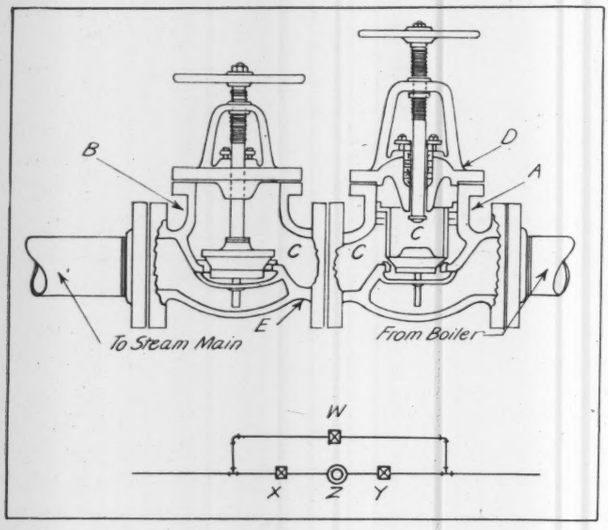
Cases of a similar kind occur when compressed air is used for testing. It is not advisable to use high-pressure air or steam for testing standard-weight cast-iron radiators, vessels, containers and similar equipment, unless proper precautions are taken to prevent an excess pressure accumulating. Applying a pressure gage and closing off the air supply valve when sufficient pressure for testing has accumulated in the appliances to be tested is not always safe practice. A slight leakage through the valve will frequently cause the pressure to build up quickly and get beyond control. In addition to a reliable pressure gage, a suitable relief valve should be placed in the testing connection, between the air or steam supply valve and the appliance to be tested. The relief valve will relieve excess pressure in case the supply valve leaks, fails to seat properly or is carelessly left open.

Safe working practice should be observed when removing valve bonnets and cap flanges from reducing valves, boiler header and blow-off valves, check valves and any other equipment that is closed off for dismantling when the pressure exists on the connecting lines. Steam pressure should be carefully considered, as workmen have been burned badly when engaged in repair work of this kind.

Referring to the drawing of the boiler steam-header valves, when a boiler is not in use the bonnet may be removed from the valve *A* by closing the valve *B*, as steam still remains on the steam main. A slight leakage of steam through the seat of valve *B* will collect in the interior space *C* of both valves. When the bonnet is removed steam will blow out and may injure workmen.

A common practice is to loosen the bolts of the valve bonnet *D* and raise it carefully, allowing the steam to escape. A safer way is to provide the valve body with a tapped opening in this space and put in a small valve that can be opened and allow the steam to escape to the atmosphere, relieving the pressure. This gives positive assurance that there is no stored up energy to cause trouble and affords a quick and convenient way of testing the valve nearer the steam main for leakage. This

Safety in the plant is more than a matter of proper use of ladders, wearing goggles, putting guards on equipment, etc. By giving a little thought to the arrangement of piping from a safety angle, much can be done to reduce potential hazards to life and limb, costly to management and worker alike. It's one of Mr. Wilson's pet subjects; his article describes two "safety piping measures" from his own experience in a large plant.



tapping should, preferably, be on the lower part of the body of the valve, as at *E*, if it does not interfere with the flange-bolt nuts, so that hot water from the steam leakage will not collect in the valve. For valves 6-in. and larger, a ¾-in. tapping is suitable.

This practice can be used to advantage for piping connections to reducing valves, or any equipment where a section of a pipe line is to be closed off for inspection or repairs and pressure is still maintained on the rest of the system of piping, as explained by the small diagram. When the valves *X* and *Y* are closed to permit the reducing valve *Z* to be taken apart for repairs, the by-pass valve *W* is opened, to keep the pipe line in service. This allows pressure to remain in the piping up to the valves *X* and *Y*. The pressure remaining in the reducing valve, together with any leakage through the valves that are closed, must be relieved. The small valve mentioned before is a safe way of doing this.

Above—Steam-header valves; a slight leakage of steam through the seat of Valve B will collect in the Space C and will blow out when the bonnet is removed. A tapped opening at E with a small valve can be used for allowing the steam to escape. Below—A small valve for relieving pressure can be used to advantage in a lay-out such as this.

Publishes Book on Calculating Heat Transmission

"THE Calculation of Heat Transmission," by Margaret Fishenden and Owen A. Saunders, has recently been published by H. M. Stationery Office, Adastral House, Kingsway, London W. C. 2. This 292-page book contains over fifty illustrations, thirteen tables, and is divided into ten chapters and three appendices. A complete bibliography is given, as is a subject and author index.

In compiling the book, it was the object of the authors to bring together in easily available form the results of experimental investigations of heat transmission, interpreting and comparing them in the light of the fundamental principles of radiation, conduction, and convection. Numerical examples are given showing the application of available information.

The chapter headings are: Radiation, Calculation of Radiation Heat Transfer, Emission and Absorption of Radiation by Gases and Flames, Conduction, Calculation of Conduction Heat Transfer, Convection, Natural Convection in Gases, Convection in Liquids, and Heat Transfer Calculations. The three appendices are: The Principle of Similarity, Proofs of Formula for Parallel and Counter Flow, and Physical Constants for Gases and Water.

Fairchild Aerial Surveys, Inc.

42,800 ft. of underground steam mains installed in approximately 19,000 ft. of tunnel and conduit will supply over a score of public buildings in Washington, D. C., with district steam. The distribution system is designed for a 300-lb. working pressure. The majority of joints will be welded and all welds in the piping will be annealed in special furnaces. Part of the section to be served by the system is shown here.

INTERESTING Equipment..... Developments

Two-Ply Steel Announced

A TWO-PLY, stainless steel, which among other uses is adapted for heating and air conditioning installations, has lately been announced. The manufacturer is the Ingersoll Steel & Disc Co., Chicago, a division of the Borg-Warner Corporation.

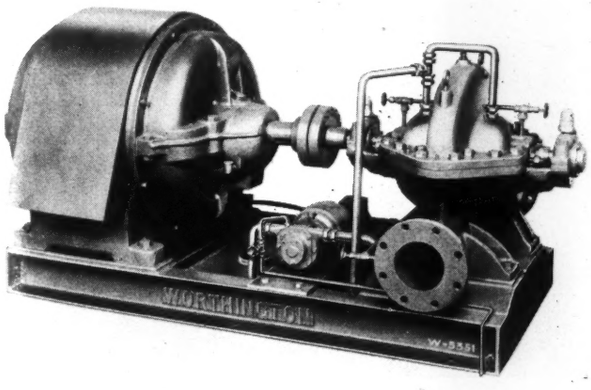
This metal gives a stainless-steel face and a carbon-steel back; it is rolled from one ingot. Its producers plan its use in such work as heating and ventilating ducts, fans, unit heater and air washing and humidifying apparatus. The steel can be deep drawn, stamped, welded, formed and polished.

Pump Is Self Priming

ANNOUNCEMENT has been made by Worthington Pump and Machinery Corporation, Harrison, N. J., of a new self-contained automatically-primed centrifugal pumping unit. It comprises an electrically-driven ball-bearing centrifugal pump, mounted with its motor on a fabricated steel bedplate, together with a priming unit of the wet vacuum type, controlled by an electric pressure switch.

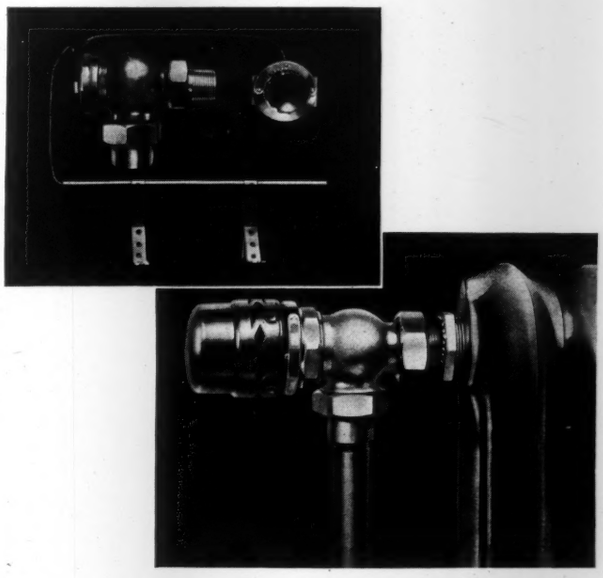
The evacuator is connected to the "high spot" of the suction volute by means of tubing, and is sealed with clear water held in a reservoir built into the bedplate of the complete unit.

The evacuator motor is connected across the main pump motor line so that in starting the motor the evacuator motor also starts. The pressure switch is placed in the evacuator motor circuit and its pressure connection is piped to the discharge nozzle of the main pump. Whenever the discharge pressure of the main pump is below a pre-determined point the pressure switch is held in the closed position. When the discharge pressure comes up to normal the switch opens and remains open as long as the pressure is maintained. If the pump loses its prime the discharge pressure naturally drops. The pressure switch is thereby closed, causing the evacuator to operate until the pressure is again built up to normal. A check valve in the evacuator suction line prevents air leakage from the evacuator to the main pump when the evacuator is not in operation.



Valves Control Temperature

THE illustrations show radiator valves which recently have been placed on the market for individual room temperature control. The lower illustration shows



the valve for exposed radiation. It may replace the ordinary radiator valve in many cases, and therefore is adaptable to existing buildings. The thermostatic mem-

ber is a liquid-filled bellows which is constantly exposed to the air of the room, but is insulated from the valve body. The housing is designed to prevent tampering. Temperature range is set at the factory from 60 to 90 F and intermediate temperatures may be obtained by turning the valve head. Once set, the head may be locked against tampering or accidental change. The 60-F low limit is designed to guard against freezing.

The model at the upper left is for use with enclosed radiation. Control is provided at two points by two bulbs separate from the steam valve proper at the cool air inlet at the bottom and at the heated air outlet at the top of the enclosure.

The manufacturer is the Fulton Sylphon Co., Knoxville, Tenn.

Gasket Resists Oil

OIL at a high temperature will not affect gaskets of a new substance that has been developed in the General Electric research laboratory, Schenectady, N. Y. The exposed edge of the gasket is not attacked, nor does the oil penetrate it. Oil-filled assemblies have been operated on test at from 210 to 230 F for a year without effect on the gasket and without leaks.

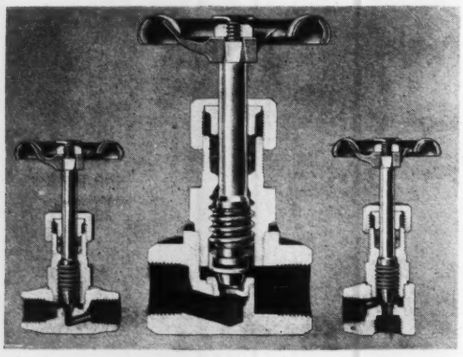
The compound may be used in contact with cemented joints; dilute acids or dilute alkaline solutions do not affect it. The material is a gray or brown, odorless and sulphur-free alkyd resin material.

Bushing assemblies of the compound with porcelain, brass, cast iron and copper have withstood alternate heating and cooling while under pressure and in contact with oil.

Among applications recommended are those where resistance to hot oil is of primary importance, where exposure to naphtha, gasoline, kerosene, benzine and similar solvents may occur and where moisture is to be excluded and oil resisted.

Valves for Close Control

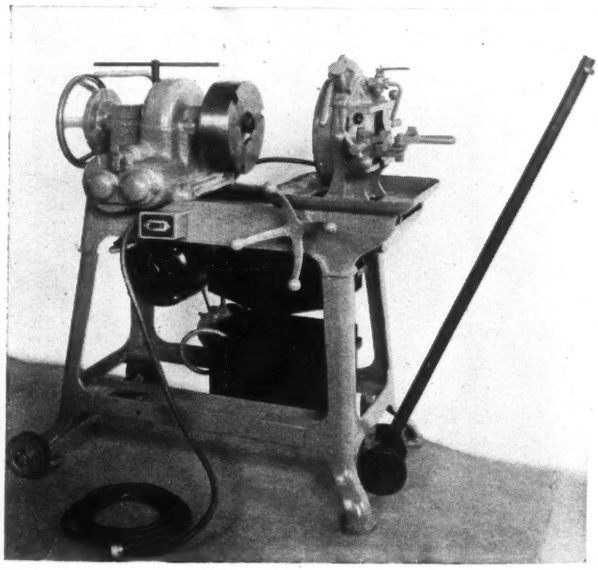
RECOMMENDED for services where high strength and close control are desirable are the ¾- and 1-in. globe valves which are an addition to the Reading-Pratt & Cady Company, Inc. (Bridgeport, Conn.), line of bar stock valves. The valves are precision turned from carbon steel and have stainless steel, slip-on renewable discs and stainless steel screw-in seats. Stems are designed and treated for temperature service and all other parts are protected against corrosion.



Smaller sized carbon steel valves, in both the globe and angle patterns, have integral seats and discs—otherwise they are similar to the larger valves. Working pressures are 4,000 lb. at 150 F, 600 lb. at 500 F. Bronze valves for 500 lb. pressure at 150 F are made in the globe, angle, and cross patterns in sizes up to ½ in. The stainless steel globe and angle valves are made in sizes up to 1 in. and for pressures up to 4,000 lb. at 150 F, or 600 lb. at 500 F. The valves can be furnished with indicator handwheels, lock shield and key, male or female unions, and male and female ends. There is also a manifold type consisting of three valves on a single body.

Die-Head Is Automatic

THE illustration below shows a die-head which has been designed to be automatic, self-opening and quick setting. The entire machine is portable and is for pipe and bolt threading. It requires two sets of dies to thread all sizes of pipe from ½-in. to 2-in. inclusive; one for ½-in. and ¾-in. and the other for 1-in., 1¼-in., 1½-in., and 2-in. The die-controlling cam ring carries a notched segment, each notch in



which corresponds to and is marked for a given setting of the dies. The notched segment is engaged by a pawl so that, as the cam ring is rotated under it, it engages first one notch and then the next and the next, until the correct one is reached and the pawl, engaging it, locks the dies in the exact position required to produce that size of thread. When a standard length of thread is to be cut, a spring plunger is depressed before the dies are set. Then, when the full length thread has been cut, the plunger trips the pawl, the dies open and the pipe is free to be removed from the machine.

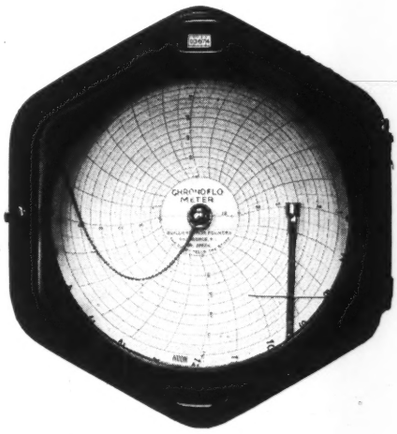
This item is manufactured by American Die & Tool Co., Reading, Pa.

Flow Meter for Many Uses

A CHART record reliable to 1/12th the maximum rate and a total flow register accurate to 1/20th of the maximum flow makes the new flow meter announced this month by the Builders Iron Foundry, Providence, R. I., particularly applicable to steam, air, gas, and liquid lines where there are wide seasonal, day and night, or hourly variations in the rate of flow. The instrument may be used with either a venturi tube or an orifice plate.

It consists of a telechron-driven contact device controlled by the height of the mercury column in the mercury unit (i. e. the rate of flow through the orifice or venturi tube) and a telechron-driven register-recorder which is in operation during the exact time that the mercury switch is "on" in the mercury unit. A power relay transmits the contact engaging the telechron drive with the integrating and recording movement through an electro-magnetic clutch.

Other features include the hexagonally-shaped case allowing the use of a long side hinge which, in turn,



allows the mounting of all the working parts on a single plate which can be swung out from the case for inspection.

Easy-to-Install Superheater

A NEW and improved superheater was developed recently by The Superheater Company, 60 E. 42nd St., New York City, which may be installed readily by regular plant attendants in h.r.t. boilers. No changes are required in settings or arches of the boiler or in other plant equipment.

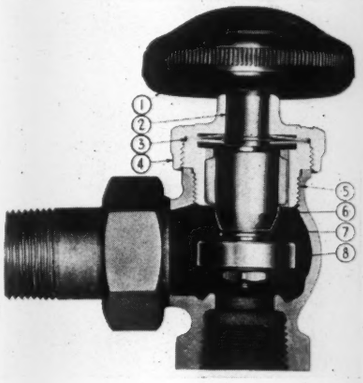
This superheater consists of two cast-steel headers, suspended one on each side of the boiler shell which are connected by detachable tubular elements or units extending around the underside of the boiler. The headers are supported by adjustable clamped steel rods. By raising or lowering the headers, the units are correspondingly raised or lowered in the combustion chamber, which regulates the final temperature of the steam.

Operation is automatic. Saturated steam is carried from the boiler nozzle through a connecting pipe to one header, known as the saturated-steam header. From there it passes to the other header or superheated-steam header, through the units under the boiler, during which it is subjected to the relatively intense heat of the gases of combustion in the combustion chamber. The steam, then superheated, passes to the main steam line for use in the plant.

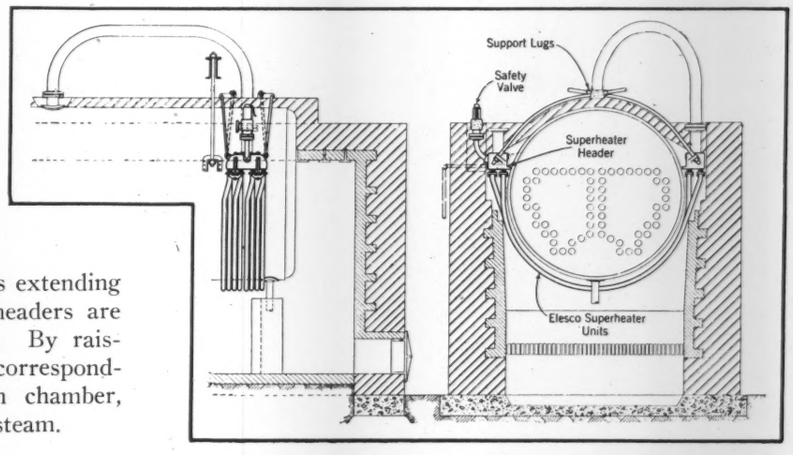
By using superheated steam it is often possible to obtain economies in the plant.

Bonnet Assembly for Radiators

THE bonnet assembly illustrated here is a replacement unit designed to improve the operation of radiators. It is operated by the valve stem which is

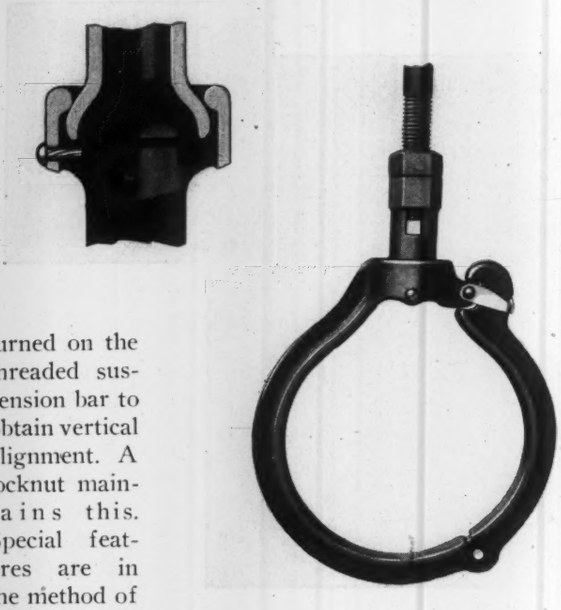


threaded into the disc-holder nut. Rotation of the valve handle causes the disc to be raised or lowered in the body of the valve, thus regulating the flow of steam into the radiator. The sealing discs (No. 3) consisting of two sets of thin metal discs, form the packless sealing arrangement of this valve. The assembly is so constructed that it fits almost any standard make and size of packed or packless inlet valves whether the bonnet threads are on the outside or the inside of the valve body. Jas. P. Marsh Corporation, 2073 Southport Ave., Chicago, makes this assembly.



A Ball-Swivel Pipe Hanger

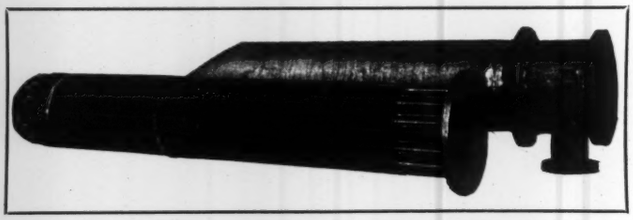
THE ball and socket joint of the pipe hanger shown here provides flexibility and allows the ring to swing in any direction, in order to compensate for expansion and contraction of the line and irregularities in alignment. The construction permits the swivel to be turned on the threaded suspension bar to obtain vertical alignment. A locknut maintains this. Special features are in the method of supporting the load and in locking the hinged arm.



The Grabler Mfg. Co. of Cleveland, Ohio, is the manufacturer.

Heater for Oil Tanks

A STORAGE tank oil heater recently placed on the market by the Griscom-Russell Company, 285 Madison Ave., New York City, is distinguished by patented heating elements which consist of a number of U-shaped steel pipes having longitudinal fins along the exterior surface of the straight portions of each pipe. These fins are equally spaced about the circumference of the pipe, and are securely fastened by a mechanical process. The purpose of the longitudinal fins is to increase the heat-transmitting surface on the outside of the pipes in contact with the oil to be heated for withdrawal from the tank. Area of the fins on a pipe is from 4½ to 6 times greater than that of a corresponding bare pipe.



The heater is installed with the shell and heating elements extending into the tank and the steam, drain and oil connections outside the tank. No cross baffles are necessary in the shell.

Air and Control Valve

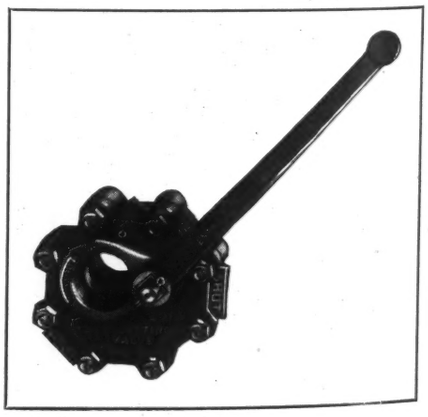
A COMBINATION air valve and room-temperature control valve which can be set at any point between 60 and 80 F has been announced by The Dole Valve Company, Chicago, Ill. Its operation is as follows:

Suppose the dial is set at 72 F. As soon as the room temperature reaches 72, no more steam is permitted to enter the radiator; extra steam is "blocked off" by a barrier of air inside the radiator. As time passes the temperature in the room begins to drop, part of the barrier of protective air is thermostatically released from the radiator in order to permit the entry of more steam, until the temperature again becomes the desired 72 F. This constant adjustment is automatic and it maintains the temperature as set on the dial.

In installing the valve, it is substituted for the air or vacuum valve. Working parts are of durable metal, while the outer casing is made of a plastic molded product.

Valve Disc Rotates

THE principle of operation of a recently-improved valve which is recommended for use on terminals where tightness is essential and for process work where easy operation is desirable is the passing of a disc over the outlet port of the valve to effect a seal. The contact surface of the disc and the entire surface of the seat and extension face



are ground together; the disc when moving across the outlet bonnet of the valve has a rotating motion to regrind one to the other.

The disc is held in position by a spring. Operation is accomplished by means of a wrench which transmits motion to a lever arm. Flow is straight through the valve and the outlet orifice is larger than the inlet orifice to permit a column of water to pass into the outlet without impinging on the sealing face around the outlet to minimize erosion of the sealing surface when full open.

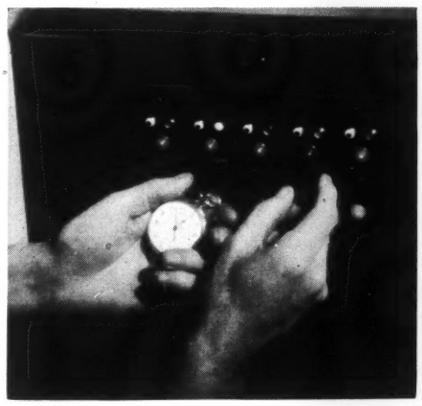
Improvements stressed by the manufacturer, Everlasting Valve Company, 1 Montgomery St., Jersey City, N. J., include a new post spring, the use of two post gaskets and a square head and wrench.

New Supervisory Control

SPEED of a new type of supervisory control permits selection and operation of the control circuits of any piece of remotely-controlled apparatus in less than one second. This is accomplished by a principle new to supervisory control known as "direct selection."

This method of control is available for use on two and four line wires. The two-wire type controls up to 25 remote apparatus units and the four-wire type controls up to 2500 units—all at the same high speed. Both the two- and four-wire types permit a number of remote stations to be operated on the same two or four wires. In addition, remote metering and remote synchronizing may be obtained without the use of any extra line wires.

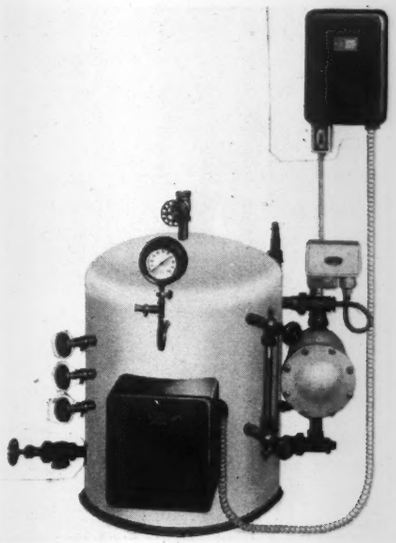
The method is suitable for applications such as the control of any type or size of remote station on power systems, electric railways, electrified railroads, oil-line



pumping, etc. The illustration shows the control's speed being checked. Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa., is the maker.

Supply Process Steam

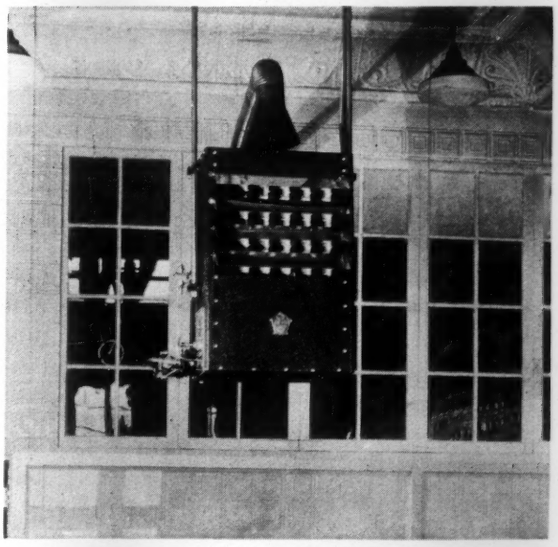
ELECTRICALLY-HEATED full-automatic steam boilers for producing small or moderate quantities of process steam at from 1 to 200 lb. pressure have been announced by the Commonwealth Electric and Manufacturing Company, 93 Boston St., Boston, Mass. The six standard models have ratings of 7½, 15, 30, 60 and 90 kilowatts; ¾, 1½, 3, 6 and 9 boiler-horsepower; or 22, 45, 90, 135, 180 and 270 lb. of steam per hour.



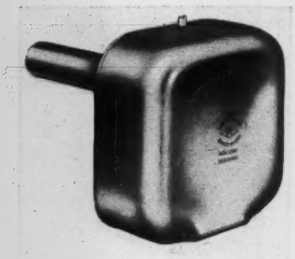
Such boilers are particularly applicable to situations which arise in many industries where small quantities of process steam are required at times when the main boiler plant may be shut down, where a small amount of steam may be required at a higher pressure than that at which the main boilers operate, steam needed at an isolated point, etc. They are constructed to raise clean, dry steam one-half to three-quarters of an hour from starting, providing a reserve quantity of steam to handle sudden large demands.

Units Are Gas-Fired

THE suspended-type gas-fired unit heater illustrated below was especially designed for heating small stores and similar areas where space is at a premium. The units have cast-iron radiation with only two joints in the heating surface. Stand-by temperature is a feature, according to the manufacturer (F. O. Schoedinger, Columbus, Ohio) enabling quick delivery of heat when the fan is started.



Controls Liquid Temperature



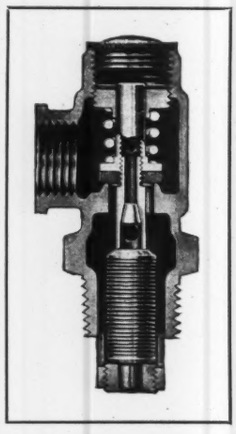
THIS direct - break thermostat is designed to control the temperature of liquids in the range from 120 to 200 F. While used principally in connection with water-heating tanks, it has many other applications.

The thermostat is screwed directly into the tank, which is tapped for a ¾-in. pipe thread. A bronze casting screwed into the tank enables removing the thermostat without emptying the tank. To set the temperature, the cover is removed and the indicator put at the desired point.

The cover, which is finished in cadmium plate, is 4¼ in. by 4¼ in. by 1-11/16 in. The manufacturer is The Hart Manufacturing Company, Hartford, Conn.

Hot-Water Relief Valve

A RECENTLY improved temperature-and pressure-relief valve for hot-water heating devices has a thermostatic bellows which expands and opens the valve before the temperature reaches 212 F; when enough hot water is relieved to allow the temperature to drop 10 degrees the bellows contracts and the valve closes. An added safety feature is a fusible plug which will melt should the temperature reach 245 F, proof that the valve is installed improperly or that the circulating piping is faulty.

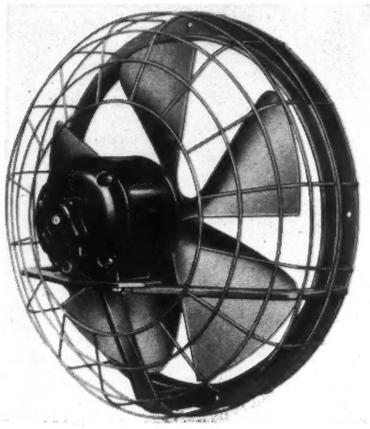


The standard setting of the temperature relief is 205 F; the pressure relief may be set at any point between 30 and 175 lb. per sq. in. by means of a slotted adjustment screw. The maker is Watts Regulator Co., Lawrence, Mass.

Feeder Easy to Install

EASE of installation is a feature of the water feeder for boilers under 2,500 ft. capacity illustrated at the right, according to the manufacturer, McDonnell & Miller, Wrigley Bldg., Chicago, Ill. The gage glass on the boiler is removed and the feeder installed in the gage-glass tappings, the gage glass then being installed on the feeder. The goose-neck is adjustable so that the feeder may be applied to any boiler within the capacity range for which it is designed.

Two bellows seal the points between the float chamber and the lever compartment and between the lever compartment and the feed valve. Valves are of stainless steel; all moving parts are away from the heat of the float chamber.



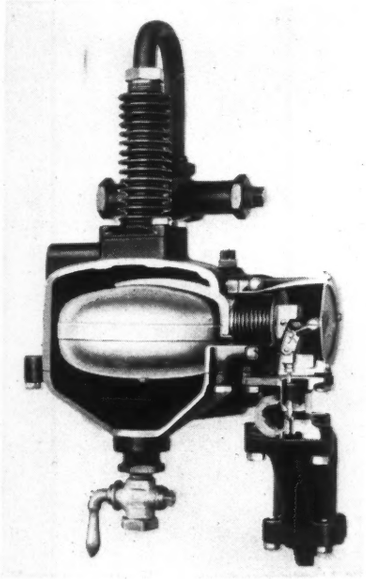
Fans Have Safety Guards

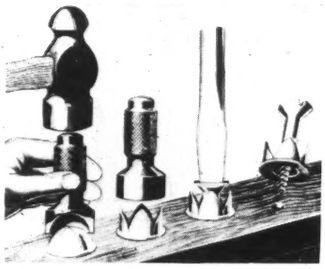
SAFETY guards of welded-steel construction are a recent improvement to a line of ventilating fans made by the L. J. Wing Mfg. Co., 154 W. 14th St., New York City, as illustrated above. Models are available for single, two and three-phase alternating current, direct current, pulley drive; high speed types for industrial use are available.

The fan wheel is of a screw-propeller type designed to give a uniform velocity across the air stream rather than a high velocity at the circumference and a low velocity in the center.

Conceals Screw Heads

A DEVICE to conceal the heads of bolts and screws is shown at the top of the next page. It is a flat washer with multiple points projecting at right angles in such a manner as to give the appearance of a crown. The crowner is placed on the shank of the screw or bolt, which is then driven home. A special tool furnished by the manufacturers is placed over the projecting points and tapped lightly with a hammer, drawing the points together over the center of the washer to form a smooth, hemispherical cap which conceals the screw or bolt head



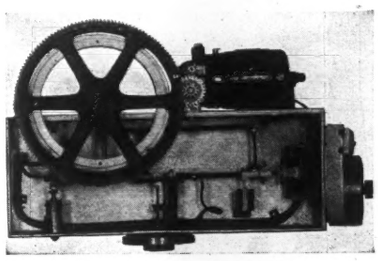


and seals it against tampering.

These devices are made of rust-proof metal to match any desired finish by The Rawlplug Company, Inc., 98 Lafayette St., New York City.

Regulates Furnace Draft

THIS electric-motor-operated furnace-draft regulator is especially designed for boilers using forced draft to control the forced-draft fan speed, or a damper between the fan and stoker, or to control both the forced draft and the stoker speed.



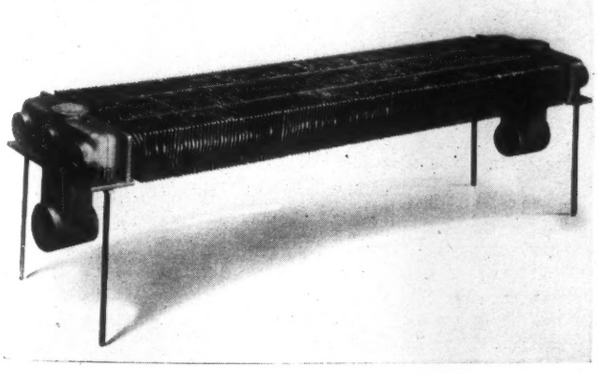
It is connected up to operate on the boiler damper when the steam-pressure control is operating on the forced draft and stoker speeds, or it can be connected up to function on the forced-draft fan speed or a damper between the fan and stoker; the steam-pressure control would then be connected to the boiler damper and the stoker engine, or rheostat if the stoker is operated by a variable-speed motor.

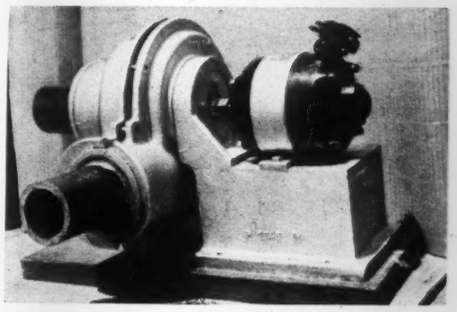
The manufacturer is the Defender Automatic Regulator Co., 308 S. 8th St., St. Louis, Mo.

A Built-in Fin-Type Heater

ANNOUNCEMENT has been made of a built-in heater of the fin type. It is shown below. The header has been designed with the idea of making it fit all standard types of valves and traps. Eccentric connections are provided for drainage purposes. The round metal legs attach to the headers and are threaded so as to permit adjustments, thus providing pitch or grade.

The manufacturer is Commodore Heaters Corporation, 11 West 42nd Street, New York, N. Y.





Exhauster of Stoneware

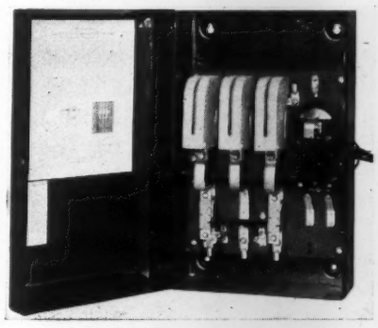
ALL the parts of the low-pressure type exhauster illustrated above which come in contact with the gases being handled are made of chemical stoneware protected by an outer armor of cast iron. The impellers of these exhausters, which are of a non-clogging type, are fastened to the motor-shafts by a four-jaw clamp, insuring proper alignment and rigidity but enabling removal. The discharge may be swung through 360 degrees.

The United States Stoneware Company, 50 Church St., New York City, is the maker.

Switches Have Thermal Overload Relays

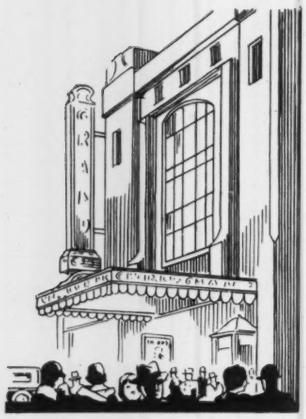
THE line of full-voltage starters for squirrel-cage induction motors made by the General Electric Company, Schenectady, N. Y., is now complete for sizes up to 1200 hp., all switches conforming to *NEMA* requirements.

The switches include new thermal overload relays which follow closely the heating characteristics of the motor. Relay heaters on the first four sizes are interchangeable as regards the mounting dimensions. Enclosing cases for the various switches are similar in appearance and all forms are available in cast iron or boiler plate case for installation in dusty or damp atmospheres; any magnetic switch in the line can be obtained in form suitable for oil immersion where the installation atmosphere is corrosive. Explosion-proof switches are available for hazardous conditions.



The line of combination switches using a magnetic switch and a fusible or non-fusible motor circuit switch has been augmented by a device rated 25 hp., 220 volts, and 50 hp., 440/550 volts. These combination switches use the same thermal overload relays that have been incorporated in the standard full voltage starters. Any of the combination switches can be obtained with test jacks, if required. All forms and sizes are available in cast iron, dust-tight enclosing cases.

"Twenty Degrees Colder Inside"



By E. Roger Hewitt

HOW many theaters display signs like this during summer or "Seventy Degrees All Year Round," instead of "Comfortable Temperatures," "No Drafts," "Fresh Clean Air," "Proper Humidity," etc.? The public complains about the frigid temperatures of many theaters and auditoriums, and is gradually rebelling against the ice-box coldness that is felt when one enters many so-called air conditioned houses. It is the topic of cartoons, slap-stick comedy, and general comment.

Various research laboratories and health departments have concluded that the temperature within a theater or auditorium, when people are at rest, should not exceed about twelve degrees below the outside street temperature, with a humidity of from 50 to 60 per cent. This is a good rule to follow if the operating engineer uses discretion in regulating temperatures to meet the outside conditions. Nevertheless, this rule does not always apply to local conditions, where the outside wet bulb temperature must be taken into consideration. Therefore, the engineer is handicapped in not knowing how to regulate his plant to obtain the proper humidity in his house.

For this purpose, the author has compiled the following tables; they were devised for the operator, and may not coincide with those for theoretical calculations. However, they have proved helpful and satisfactory for practical results.

The large table represents a relative humidity chart,

Table for Determining Proper Relative Humidity and Dew Points and Temp. of Saturated Air Leaving Washer. (Bar. Press. 29.92 In.)

Outside Dry Bulb F	Inside Dry Bulb F	Degrees Difference Between Dry & Wet Bulb Temp.															
		9		10		11		12		13		14		15		16	
		% R. H.	Dew Point	% R. H.	Dew Point	% R. H.	Dew Point	% R. H.	Dew Point	% R. H.	Dew Point	% R. H.	Dew Point	% R. H	Dew Point	% R. H.	Dew Point
70 F or Less	68	58	53	54	51	50	49	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
72	70	59	55	55	53	51	51	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
75	71	60	56	56	54	52	52	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
78	72	.....	.....	57	56	53	54	49	52	.....	.....	.....	.....	.....	.....	.....	.....
80	73	.....	.....	57	57	53	55	50	53	.....	.....	.....	.....	.....	.....	.....	.....
84	74	.....	.....	58	58	54	56	50	54	.....	.....	.....	.....	.....	.....	.....	.....
86	75	.....	.....	58	59	54	57	51	55	.....	.....	.....	.....	.....	.....	.....	.....
88	76	.....	.....	59	60	55	59	51	57	.....	.....	.....	.....	.....	.....	.....	.....
89	77	.....	.....	59	62	55	60	52	58	.....	.....	.....	.....	.....	.....	.....	.....
90	78	.....	.....	60	63	56	61	53	59	.....	.....	.....	.....	.....	.....	.....	.....
92	79	.....	.....	60	64	57	62	53	60	50	59	.....	.....	.....	.....	.....	.....
94	80	.....	.....	.....	.....	57	63	54	62	50	60	.....	.....	.....	.....	.....	.....
96	81	.....	.....	.....	.....	58	65	55	63	51	61	.....	.....	.....	.....	.....	.....
98	82	.....	.....	.....	.....	58	66	55	64	51	62	.....	.....	.....	.....	.....	.....
100	83	.....	.....	.....	.....	59	67	56	65	52	64	49	62	.....	.....	.....	.....
102	84	.....	.....	.....	.....	59	68	56	66	52	65	49	63	.....	.....	.....	.....
104	86	.....	.....	.....	.....	.....	.....	57	69	53	67	50	65	47	64	44	62
106	88	.....	.....	.....	.....	.....	.....	57	71	54	69	51	68	48	66	46	64

Following Temp. Saturated Air Based on 30 C.F.M. Per Seat.

D. P. of Air in Auditorium.....	52	54	55	56	57	58	59	60	61	62	63	65	67	68
Temp. Sat. Air—No By-passing	49	51	52	53	54	55	56	57	58	59	60	62	65	66
Temp. Sat. Air Using By-pass.....	45	47	48	49	50	51	52	53	54	56	57	59	61	62

showing only the humidities within the comfort zone. In the column at the left of the table, is shown various *outside dry bulb temperatures,* with the corresponding comfortable *inside dry bulb* temperatures in the second column. Reading to the right, the operator may select the most desirable *relative humidity,* and find the corresponding *dew point* for the air at this condition. The horizontal column across the top shows the difference in degrees between the dry bulb and wet bulb temperatures.

The smaller table was devised so the proper temperature of saturated air leaving the cooling unit or washer may be determined. There are two values given, one of which is for systems where all the air is passed through the washer, with no *recirculated air* being by-passed around the washer. The other is for a system where a quantity of *recirculated air* is being by-passed for the purpose of *tempering* or adding heat to the saturated air, after it leaves the washer. This value is necessarily lower, for the moisture content of the saturated air must be lower to compensate for the moisture in the *recirculated air* which is by-passed into the mixing chamber.

The table of saturated air temperatures is based on an air quantity of 30 c.f.m. per seat, for the evaporation load is assumed as one-half grain of moisture per cubic foot air per minute. Therefore, when the audience is less than 100 per cent the engineer must be governed by experiment in correcting these figures. If the plant has variable speed fan control, volume of air circulated may be regulated so as to kee .s close as possible to the 30 c.f.m. per person (occup. t seat) basis.

Steam Consumption of Buildings

Table Shows Amount and Cost of Steam Used by Several Types of Buildings

RECORDS of actual steam or fuel consumption of groups of buildings are of value to the engineer in estimating the requirements of similar structures. Care and judgment must be used, of course, in order that the data from one building may be applied to an estimate involving another building which has different characteristics and which is subject to different climatic conditions, occupancy, etc.

The table at the bottom of this page is published through the courtesy of Robert L. Fitzgerald, manager of the heating department of the Indianapolis Power & Light Company of Indianapolis, Ind.

The weighted average values shown in the next to the last column are determined by dividing the cost as shown in the third column by the total radiation as shown in the fourth column. This average gives effect to the difference in the amounts of radiation in the different groups. The arithmetical average is merely the average of the averages.

The total cost in the third column is the actual cost of the steam to the customer provided the bill was paid before the prompt payment discount.

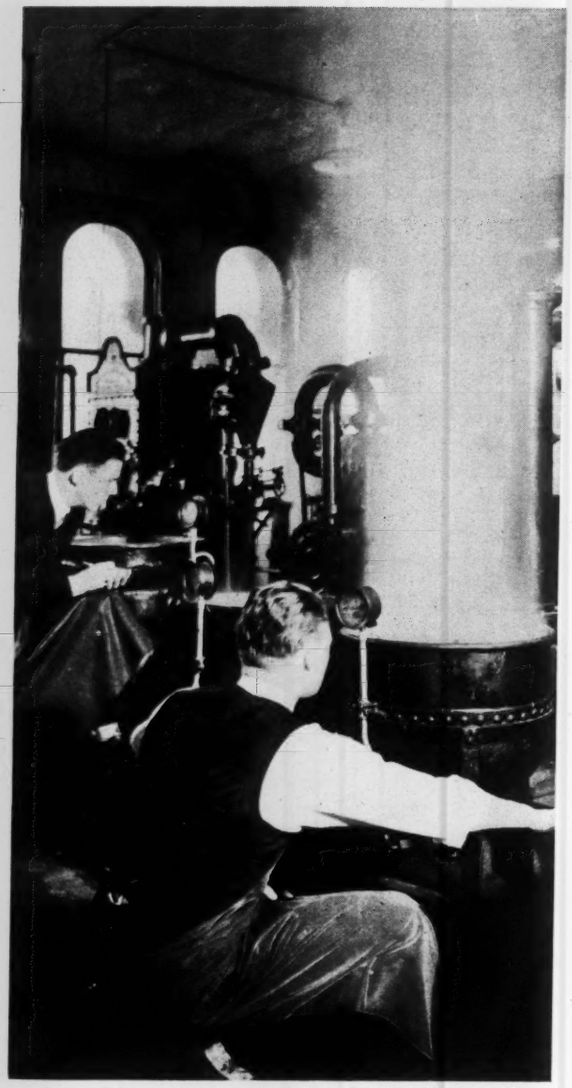
The average cost of 19½ cents as shown for public buildings—which might appear out of line with the other values—is correct and is due to the effect of the Indiana State Capitol being included in this group. This is a very large building, liberally set with radiation and with walls that average about four feet in thickness. The low heat loss through these thick walls provides a very low annual consumption for this particular building, and its size is such as to affect the weighted average of the group.

The average rate per 1,000 pounds of steam for the entire group is 68.6 cents per thousand pounds.

Average Steam Consumption of Various Types of Buildings in Indianapolis Using Central Station Steam

1930-31 *Season. Average Mean Temperature,* 44 *F*

Buildings	No.	Total Steam Consumption Lb.	Total Cost	Total Radiation Sq. Ft.	Lb. Steam Per Sq. Ft. Radiation (Av.)	Cost in Cents Per Sq. Ft. Radiation	
						Weighted Av.	Arith. Av.
Hotels	11	44,898,600	$31,133.41	93,047	482	33	37
Apartments	13	37,660,400	26,058.90	71,415	526	36½	39
Dept. Stores	6	34,721,500	23,527.15	110,891	315	21	24
Theaters	5	21,109,500	14,532.85	58,077	364	25	37
Office Buildings	20	94,065,200	64,366.85	265,562	354	24	25
Auto Sales	8	27,879,700	19,359.85	55,826	498	35	38
Public Garages	3	6,178,500	4,442.35	22,013	280	20	24
Manufacturing	4	17,990,000	12,370.55	40,279	447	31	31
Public Buildings	4	25,692,900	17,218.95	88,149	292	19½	33
Clubs—Lodges	3	6,384,800	4,625.50	12,743	500	36	37
Printing Establishments	5	23,811,900	16,277.15	36,352	655	44½	44
Warehouses	2	8,102,200	5,685.40	13,761	590	41	37
Mean Total	84	348,495,200	$239,598.91	868,115	400	27½	33



TWO WAYS TO PREPARE FOR TOMORROW

Heating · Piping and Air Conditioning
EDITORIAL

Two points about engineering and engineering research were picked up last month which should be interesting to the director of engineering and to the research engineer. One was overheard in a conversation and the other is from a book published by a technical institute.

THE CONVERSATION:

Banker: Before we lend you that $100,000 to extend your manufacturing facilities, we would like to inquire into the work of your Research Department.

Vice-President in Charge of Engineering: We have no Research Department.

Banker: I am sorry. We would seriously question the policy of any manufacturer who, these days, is not preparing for tomorrow by conducting research.

THE BOOK:

It is a beautiful book published by Armour Institute of Technology, *entitled "A New Approach to Engineering Education." Among other things it says that industry always pays for engineering education, anyway, so why should it not recommend the kind of education to be provided? It costs less to educate a man in school than in the plant. Engineering education today saves the manufacturers' cost for tomorrow.*

Heating·Piping
and Air Conditioning

1895

American Society of Heating and Ventilating Engineers
JOURNAL SECTION

SEPTEMBER 1932

Material in this Section prepared by Publication Committee: F. D. Mensing, *Chairman;* E. O. Eastwood and F. E. Giesecke
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Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures

By A. P. Kratz[1] (*MEMBER*) and E. L. Broderick[2] (*NON-MEMBER*)
Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory

This paper constitutes a presentation of results obtained from a study incident to the investigation in heating and ventilation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. The work is carried on in the Department of Mechanical Engineering under the direction of A. C. Willard, Professor of Heating and Ventilation and Head of the Department. Acknowledgment is due to J. R. Fellows, Instructor in Mechanical Engineering, for valuable assistance in the preliminary stages of the study.

THE determination of surface film conductances, or surface co-efficients of heat transmission for building walls, involves the measurement of the temperature of the bounding surface between the wall and the air. From the experimental standpoint such measurements are difficult to make, and some uncertainty usually exists as to whether the temperature indicated is the temperature of the actual surface or has been influenced by the temperature of the material behind the surface or of the air in front of it. Furthermore, the presence of the measuring instrument may become a disturbing element causing the surface in its immediate vicinity to assume a temperature different from that of the remainder of the surface outside of the bounds of its influence.

A study of methods for measuring the surface temperature of an iron steam chest was made by F. W. Adams and R. H. Kean.[3] They found that, with the exception of a specially constructed thermocouple compensated for heat losses by means of a small electric heating element, the minimum error was obtained with a thermocouple of fine wire, having the wires inserted in a small hole drilled about two millimeters into the surface, and held in place by a soft copper plug driven in between them, as recommended by P. D. Foote, C. O. Fairchild and T. R. Harrison.[4] Between the temperatures of 100 F and 300 F the last mentioned thermocouple read approximately 2.7 deg lower than the actual surface temperature. In this case, the cast-iron and the copper plug were electrically conducting materials and the thermocouple junction was formed across the surface between the two wires. Such a method is not applicable in the case of building walls in which the materials are usually more or less electrically insulating. The specially constructed compensated couple might be used in certain cases, but it requires manual application and could not be used where the surfaces are inaccessible or not exposed. Furthermore, the presence of this instrument would disturb the conditions under which the study of heat transmission coefficients is usually made, and would change the surface temperature from the normal value.

Two methods of installing thermocouples are commonly used for the study of surface temperatures of walls built of electrically non-conducting materials, namely, (1) to place the thermocouple, made of fine wire, on the surface and cover the junction and several inches of the leads with thin adhesive vellum, and (2) to embed the junction and several inches of the leads in a scratch made in the surface, and to seal the wires into the scratch with plaster of Paris, shellac, or some other plastic material. The wire and sealing material is worked flush with the surface. While very fine wires are preferable for both of these methods, wire as large as No. 22 B. and S. gage is sometimes used for the second.

In the case of the thermocouple placed on the surface and covered with vellum, since the wires extend the full diameter into the air, some uncertainty exists as to whether the temperature of the wire, and therefore the temperature indicated by the thermocouple, is influenced by the temperature of the air. The latter is usually quite different from the temperature of the surface. Also, the presence of the vellum disturbs the temperature of the surface, and corrections based on the conductivity of the vellum are somewhat speculative. In the case of the embedded thermocouple there is a possibility that the wire will assume the temperature of the layers below the surface, and that the indicated temperature will be different than the actual surface temperature. In both cases, if sufficient length of leads is not covered or embedded, heat conduction along the leads may affect the temperature indicated.

Some study of the use of embedded thermocouples for

[1] Research Professor in Mechanical Engineering, University of Illinois.
[2] Research Assistant in Mechanical Engineering, University of Illinois.
[3] Measurement of Surface Temperatures, by F. W. Adams and R. H. Kean. Contribution from the Department of Chemical Engineering, Massachusetts Institute of Technology, Serial No. 173, Oct. 1926.
[4] Pyrometric Practice, by P. D. Foote, C. O. Fairchild and T. R. Harrison, U. S. Bureau of Standards Technologic Paper No. 170, p. 298.

obtaining surface temperatures was made by A. P. Colburn and O. A. Hougen.[5] This study was made with thermocouples embedded in slots in condenser tubes by means of glycerol-litharge cement and made flush with the tube surface. Thus the wires were electrically insulated from the tube, but made good thermal contact. These thermocouples were used on the steam side of the tube. While these investigators found that this method gave results agreeing exactly with the actual surface temperature, the fact that the steam, the metal tube, the litharge cement, and the thermocouple wires were all media with relatively high thermal conductivities did not seem to guarantee that similar results would be obtained in the case of walls where the conductivities of the air, the wall and the plaster cement were all very low as compared with the conductivity of the thermocouple wires. A study of the use of theormocouples under the last named conditions, therefore, appeared to be warranted, and the objective in this investigation was a comparison of the readings of the thermocouples installed by methods (1) and (2), with the actual temperature of the undisturbed surface uninfluenced by the presence of the thermocouples themselves.

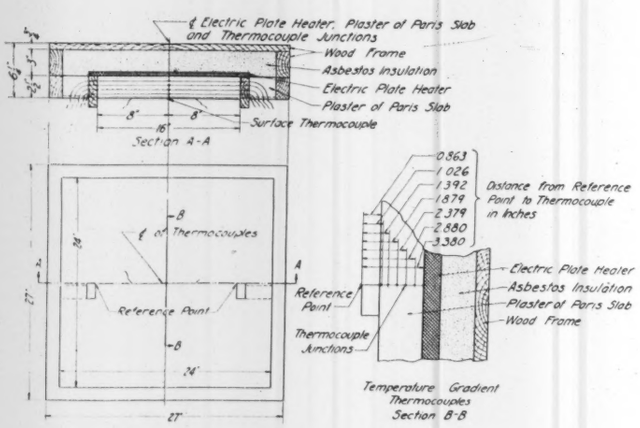


Fig. 1—Test slab.

The determination of the true surface temperature was based on the premise that if a temperature gradient curve for a known homogeneous material could be established to within ⅛ in. of the surface, this curve could be extrapolated to the surface and would accurately represent the temperature that the undisturbed surface would assume, provided that any local disturbance did not extend into the material as far as the position of the last temperature used as a point on the temperature gradient curve.

Description of Apparatus

The principal apparatus used consisted of a homogeneous plaster of Paris slab 24 in. x 24 in. and 2½ in. thick. This slab was placed in a vertical position with one surface exposed to the air of the room as shown in Fig. 1. Five thermocouples, of No. 34 B. and S. gage copper and constantan wire, were embedded in the slab with their junctions placed in the short central axis and with the lead wires in planes parallel to the faces of the slab. The distances between these junctions measured along the short central axis and perpendicular to the face of the slab were accurately known. The location of these couples and the distances between junctions are shown in the cross-section in Fig. 1. Since the method used was based on the determination of a uniform temperature gradient, the actual conductivity of the slab was not important and the effect of the thermocouple wires on the conductivity was of no consequence, provided that they did not cause irregularities in the temperature gradient.

The slab was poured in a frame in which the thermocouples had been stretched. This frame also contained two reference points across which a straight edge could be placed and the distances of the thermocouples were accurately determined with reference to the straight edge. In constructing the slab, the frame containing the thermocouples was placed on a plane surface with the two reference points up. The plaster of Paris was then poured without disturbing the thermocouples, and the upper surface was struck off with a straight edge drawn across the upper edges of the frame. After setting, this surface was sanded so that it presented a smooth plane face. By using a straight edge across the reference points, the position of the exposed slab surface relative to the thermocouples could be obtained at any time. The accuracy of the results depended to a large extent on the accuracy with which these distances could be determined.

An electric plate heater was placed against the other surface of the slab, with a thermocouple between the heater and the surface to serve as a control. This heater was insulated, so that most of the heat generated passed through the slab, and was provided with a rheostat and ammeter for adjusting and measuring the current. The electromotive force of the couples was measured by means of a precision potentiometer and all thermocouples were calibrated before installation.

Test Procedure

The three thermocouples used for indicating the surface temperature were installed and tested separately. The first thermocouple was made from No. 34 B. and S. gage copper and constantan wires with the junction filed to approximately the same diameter as the wires. It was laid on the surface, as shown in Fig. 1, and the junction and about 3 in. of the leads on each side of the junction were covered with a strip of thin adhesive vellum pressed tightly to the wire and the surface. When this series of tests was completed, the vellum was removed and a fine scratch was made in the surface. The junction and about 3 in. of the leads of this same thermocouple were then pressed into the scratch, sealed in with plaster of Paris, and worked flush with the surface. After this series of tests was completed, the same procedure was followed with a thermocouple made from No. 22 B. and S. gage copper and constantan wires. In this case, however, the junction and about 4 in. of the leads on each side were embedded to a depth of slightly more than one-half of the diameter of the wire, and, after the plaster had set, the protruding portion of the wire was filed flush with the surface. In all cases, the junction was placed on the axis of the slab and in line

[5] Studies in Heat Transmission, by A. P. Colburn and O. A. Hougen. Bulletin of the University of Wisconsin Engineering Experiment Station Series No. 70, p. 69. Also *Industrial and Engineering Chemistry*, Vol. 22, May 1930, p. 522.

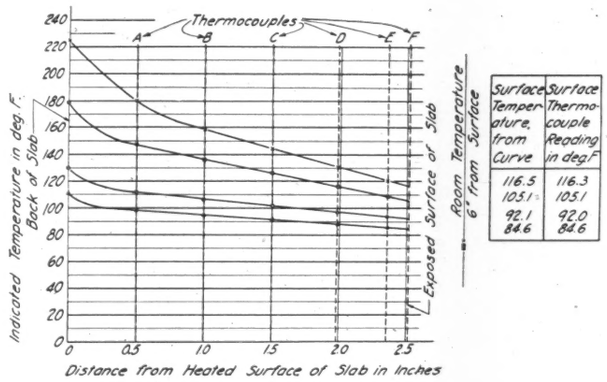


Fig. 2—Temperature gradient in plaster-of-paris slab.

with the thermocouples located inside of the slab.

All tests were run with room air temperatures between the limits of 65 and 85 F. The temperature at the back of the slab, as indicated by the thermocouple placed between the plate heater and the slab surface, was brought to some predetermined value and was maintained constant until thermal equilibrium had been established. This required from one to four hours. When thermal equilibrium had been established several observations were made of the temperatures indicated by the thermocouples located inside the slab, on the surface, and in the air about 6 in. in front of the surface.

The observed temperatures as indicated by the thermocouples located inside of the slab were then plotted on a graph in which the abscissae represented the location of the thermocouples relative to the surfaces of the slab, as shown in Fig. 2. Since the surface temperature desired was that of the undisturbed surface and not that produced and measured by the surface thermocouple, this temperature was represented by the intersection of the temperature gradient curve with the line *F* in Fig. 2 indicating the location of the exposed surface of the slab, provided that any disturbing influence caused by the presence of the surface thermocouple did not extend to the last thermocouple, *E*, located 0.163 in. below the surface. If the surface thermocouple truly indicates the surface temperature and causes no disturbance at the surface, then the reading of this thermocouple will be the same as that given by the intersection of the lines on the graph. If on the other hand the surface thermocouple

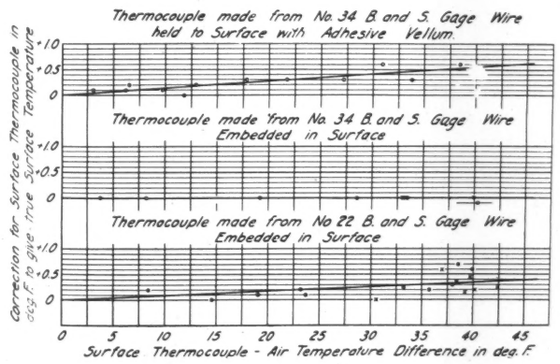

Thermocouple made from No. 34 B. and S. Gage Wire held to Surface with Adhesive Vellum
Thermocouple made from No. 34 B. and S. Gage Wire Embedded in Surface
Thermocouple made from No. 22 B. and S. Gage Wire Embedded in Surface
Correction to be Applied to Thermocouple in deg. F. to Give True Surface Temperature
+.10
+.05
0
Surface Thermocouple - Air Temperature Difference in deg. F.

Fig. 3—Surface thermocouple correction curves.

does not indicate the true surface temperature, or produces some disturbance tending to alter the true surface temperature, the difference between the temperature indicated by the surface thermocouple and that read from the intersection of the lines on the graph represents the correction to be applied to the reading of the surface thermocouple in order to obtain the true surface temperature. This procedure was followed in obtaining the corrections, and the correction was regarded as positive if the reading of the surface thermocouple was lower than the true surface temperature.

Results of Tests

Sample temperature gradient curves are shown in Fig. 2, and the correction curves for the three types of surface thermocouples used are shown in Fig. 3. In the latter figure the corrections have been plotted against the difference in temperature between the exposed surface of the slab and the air, since it was not possible to maintain constant air temperatures for all tests. It was found that the actual air temperature had only a slight influence on the magnitude of the corrections over a range of about 20 deg.

It may be noted from Fig. 3 that for all practical purposes the thermocouple made from No. 34 B. and S. gage wire and embedded in the surface indicated the true surface temperature. This same thermocouple placed on the surface and covered with thin vellum indicated a temperature lower than the true surface temperature. At a temperature difference of 40 deg between the surface and the air the correction was approximately +0.55 deg. The large couple made from No. 22 B. and S. gage wire and embedded in the surface also indicated a temperature lower than the true surface temperature, the correction amounting to +0.35 deg at a temperature difference of 40 deg between the surface and the air.

Since the large thermocouple was embedded rather deeply into the surface it seemed reasonable to expect that its reading would be influenced by the higher temperature layers just below the surface, and, if anything, that the indication of this couple would be too high, thus making a negative correction necessary. The results, however, indicated a positive correction. It was considered possible that the plaster seal might be faulty, thus allowing air to diffuse behind the thermocouple and lower the reading. Accordingly, the thermocouple and adjacent surface were painted with several coats of lacquer to act as a seal. These points are distinguished as crosses on the correction curve in Fig. 3. From this it may be noted that no difference in the results obtained by the two methods of sealing is evident.

A reasonable explanation for the performance of the thermocouple made from No. 22 B. and S. gage wire may be based on the fact that, since the conductivity of the metal is relatively high, the heat loss by conduction through the metal wires would be greater than that through the same amount of plaster of Paris. Hence, the temperature of the wire and of the surface in the immediate vicinity of the thermocouple would be lowered over that of the remainder of the surface. The couple would indicate its own temperature correctly and would, therefore, indicate a value less than the true temperature of the undisturbed surface as obtained from the graph in

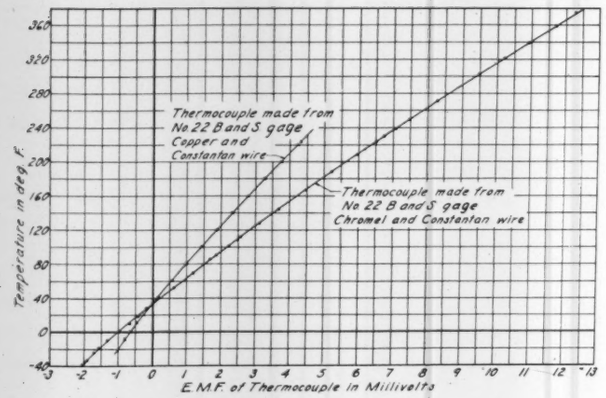

Thermocouple made from No 22 B and S gage Copper and Constantan wire
Thermocouple made from No 22 B and S gage Chromel and Constantan wire
Temperature in deg F.
E.M.F. of Thermocouple in Millivolts

Fig. 4—Thermocouple calibration curves.

Fig. 2, thus making a positive correction necessary, if the temperature disturbance did not extend in as far as couple *E*. The bright metal wires would tend to lose heat less rapidly by radiation than the surface, but the effect of the conduction loss evidently more than offset this tendency to increase the surface temperature at the thermocouple.

In the case of the small thermocouple embedded in the surface, either the mass of the wire was not sufficient to lower the temperature of the surface adjacent to the thermocouple or the amount that it was lowered was just offset by the amount that the reading of the couple would tend to be increased by the higher temperature of the layers just below the surface. In the case of the small couple placed on the surface and covered with thin vellum, the presence of the vellum would tend to raise the temperature of the surface covered. However, the thermocouple and the vellum covering it extended out into the air which was at a considerably lower temperature than the surface, and the net result was to cause the indication of the thermocouple to be lower than the true surface temperature.

For certain types of work it is sometimes advantageous to use thermocouples giving a higher electromotive force per degree temperature change than the commonly used, copper-constantan, iron-constantan, or chromel-alumel combinations. While the chromel-constantan combination is not novel, its use is rarely mentioned. Therefore, it seems worthy of some attention in connection with any study of thermocouples. In order to determine the characteristics of the chromel-constantan combination, three thermocouples were made from chromel and constantan wire and calibrated over a range of from —39 F to + 374 F. The calibration curve for one of these is shown in Fig. 4, together with the calibration curve for a copper-constantan couple to serve as a comparison. From these curves it is evident that the chromel-constantan couple is much more sensitive than the copper-constantan. For a given temperature the former generates an electromotive force approximately 50 per cent greater than the latter, and over the range tested no regions of discontinuity or inconsistency were found. Some trouble was experienced in forming the junctions, however, because these two metals are difficult to solder or fuse together.

The following conclusions may be drawn from the results of these tests as applied to the use of thermocouples in connection with materials electrically insulating and having low thermal conductivity:

1. A thermocouple made of wire as fine as No. 34 B. and S. gage having the junction and 3 in. of the leads embedded in the surface and sealed flush with the surface with cement or plaster of Paris, for all practical purposes indicates the true temperature of the surface.
2. A thermocouple made of wire as large as No. 22 B. and S. gage having the junction and 4 in. of the leads embedded in the surface and sealed flush with the surface with cement or plaster of Paris indicates a temperature somewhat lower than the true temperature of the surface.
3. A thermocouple made of wire as fine as No. 34 B. and S. gage placed on the surface and having the junction and 3 in. of leads covered with thin vellum indicates a temperature lower than the true temperature of the surface.
4. Thermocouples made by the combination of chromel and constantan wire give electromotive forces 50 per cent greater than the electromotive force given by a copper-constantan thermocouple at the same temperature.

Analytical Calculation of Complicated Air-Duct

A complete work in the Russian language under this title has been printed with 26 tables and 210 pages of text. In an English reprint of 54 pages the author, P. N. Kamenew (Stalino, Donbass, Ukraine, S.S.R.) presents two methods of computation which produce the same apparent result. The first method is by equivalent orifices and is similar to the method of blast area which is still used to some extent. The second method is simpler and is called the method of requalents and is similar to the resistance method. In either case a subsequent computation is made to correct the diameters and velocities for the fact that they will not be as originally assumed.

The author limits the method to industrial ventilation where small local resistances occur and to systems of round ducts. By use of Bernoulli's theorem a static pressure regain is obtained as the velocity along the duct system is lowered; this is known to exist. (See *Fan Engineering*, 2nd Edition, p. 54, also pp. 67-69.)

The coefficient of friction is assumed as varying with diameter only, according to Dr. Blaess. This disregards the effect of velocity, which is also important. The variation is considerably less than the value generally accepted.

The author shows a peculiar injecting effect or influence of one jet upon another at junction points. It is true that in a pressure system there is a possibility of producing a static pressure within the system less than atmospheric and obtaining an injection or inflow of air. The author ends in stating that on an exhaust system the reverse is true and that air may enter one inlet and be forced out of another. This is fundamentally incorrect and impossible since the velocity pressure of impact can never quite equal the static pressure producing that velocity. In other words, an exhaust system in which all inlets open to the same atmospheric pressure can never have a pressure existing therein greater than atmospheric pressure, and therefore cannot have outflow. *Reviewed by R. D. Madison.*

Determining Minimum Air Velocities for Exhaust Systems

By J. M. Dallavalle[1], Cleveland, Ohio
NON-MEMBER

IT IS frequently of considerable importance in the design of an exhaust installation to know the minimum velocities necessary to transport various materials. At the present time, there are no comprehensive data available to assist the engineer in the estimation of these velocities. This paper, therefore, attempts partially to fill this need by a systematic investigation of the velocities required to move particles of a specified size and density through horizontal and vertical ducts. It relates particularly to velocities necessary to avoid settling or permanent suspension (in the case of vertical pipes) and must be distinguished from *pneumatic conveying* in which case the number of particles per unit of air volume is high.

Previous Work

Burke and Plummer[2] have proposed the following equation for the suspension of macroscopic particles in a fluid stream:

$$v = K\left(\frac{\sigma - \rho}{\rho} d\right)^{1/2} \qquad (1)$$

where

v = the velocity of the fluid in centimeters per second necessary to cause suspension
σ = specific gravity of the material suspended
ρ = the specific gravity of the fluid
d = the average diameter of the particles in centimeters
K = a constant which depends on the shape of the particles. For spherical and cubical particles, K varies from 47 to 51.

If the fluid dealt with is air, ρ is usually negligible in comparison with σ, in which case Equation 1 reduces to

$$v = K'(d\sigma)^{1/2}; \quad K' = \frac{K}{\rho^{1/2}} \qquad (2)$$

which is identical to the equation proposed by Lister[3]. Martin[4] has found Equation 2 to be in satisfactory agreement with his experiments on the pneumatic size-separation of crushed quartz particles when d is greater than 0.04 cm.

Equations 1 and 2 apply only to the suspension of particles in a fluid stream. In order to transport the particles, it is necessary to have velocities greater than can be deduced from these equations. They cannot apply directly to the problems of transport without some modifications. No attempt is made, however, to adapt them to the results on hand, although some similarity is obvious. The present data are correlated in an empirical fashion and apply only to the pneumatic transport of low concentrations of materials, such as are common in most exhaust installations.

Materials, Apparatus and Procedure

Tests were run with screened cinders, crushed carbon, anthracite and quartz, which represented a fairly wide range of densities. The sizes screened varied from particles passing a 2- and retained on a 4-mesh, to those passing a 10- and retained on a 14-mesh. In all, five sizes of each material were investigated as outlined in Table 1.

The densities of the materials were determined with a pycnometer using water. The apparent density expressed in pounds per cubic inch or per cubic foot might have simplified matters considerably for the engineer, but since voids affect such determinations to a great extent, the term has not been employed. Table 2 gives the densities of the various materials.

The set-up of the apparatus used to determine the minimum air velocities required to convey a material of stated size is shown in Fig. 1. The actual duct lengths considered for the horizontal and vertical system were 16 ft and 13 ft, respectively. Some question may arise as to the fitness of conducting the experiments on the negative side of the fan, but a few tests made on the positive side during the course of the investigation

Table 1—Sizes of materials investigated.

Screening	Average Diameter* (Inches)
Through a 2- retained on 4-mesh	0.320
Through a 4- retained on 6-mesh	0.167
Through a 6- retained on 8-mesh	0.102
Through an 8- retained on 10-mesh	0.079
Through a 10- retained on 14-mesh	0.055

*By *average diameter* is meant the arithmetic average of two screen sizes. This is not statistically correct, but it simplifies the procedure in a practical way.

Table 2—Densities of various materials.

Material	Specific Gravity
Cinders	1.08
Carbon	1.26
Anthracite	1.45
Quartz	2.65

[1] Consultant in Industrial Hygiene

[2] See Suspension of Macroscopic Particles in a Turbulent Gas Stream, by Burke and Plummer (*Journal of Industrial and Engineering Chemistry*, Vol. 20, p. 1200, 1928).

[3] See The Screening and Grading of Materials, London, 1924, p. 117.

[4] See *Chemical Engineering*, London, 1928, p. 31.7.

showed a relatively small difference in the results. It is well to point out in this connection that a method for feeding the materials during a test would have been more complicated and difficult under positive pressure.

The velocities of flow were determined with a Pitot static tube mounted at the midsection of the duct system and connected to a suitable manometer. The air volumes were controlled by means of a gate valve at the exhaust. A brass pipe ½ in. in diameter and 18 in. in length was projected along the axis of the duct and was used to feed the materials. In all tests, the suction maintained by the fan was sufficient to draw the material through the pipe and into the duct system. Materials of a given size were fed for 10-min. intervals at various velocities. That velocity was taken as the minimum at which the tendency to settle was completely eliminated. Air flows in the neighborhood of this critical velocity were used for a check. The materials were fed at a constant rate at such a concentration that the velocity per unit cross section of the duct was sensibly unchanged by their presence. No observations of the speed of the material during transport were made.

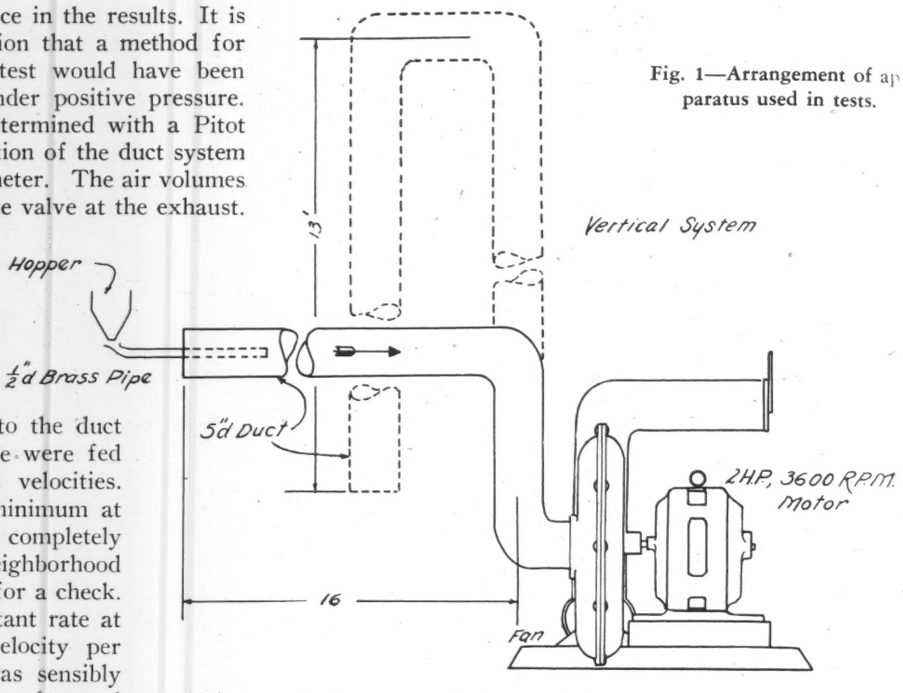


Fig. 1—Arrangement of apparatus used in tests.

The procedure of the experiments for the transport of solids in vertical ducts was identical to the one described for horizontal ducts. The general outlay of the apparatus is shown sketched in dotted lines in Fig. 1. As would be expected, the critical velocities are more definitely marked than is the case with a horizontal duct system.

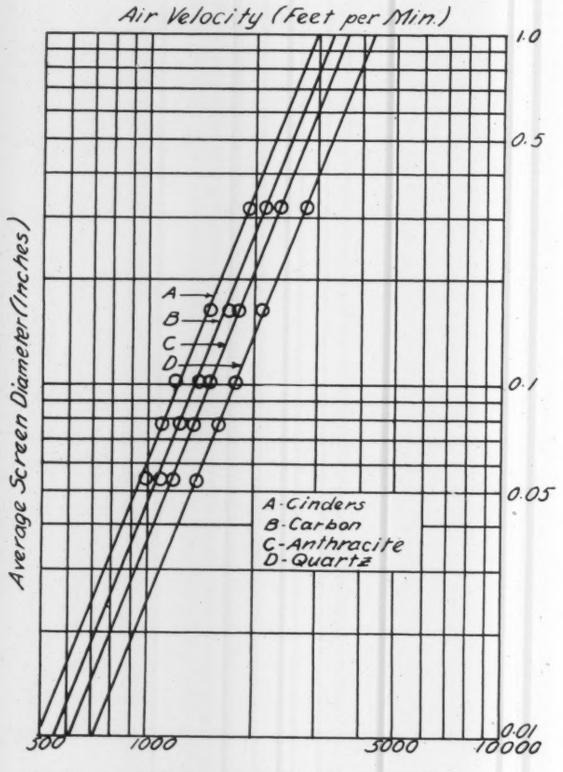


Fig. 2—Data on horizontal ducts. (Minimum velocities).

General Discussion

The experimental results obtained for horizontal and vertical transport are shown in Figs. 2 and 3, respectively. These results, from the method of plotting, indicate that the minimum velocities obey a relationship of the form

$$v = ad^n \tag{3}$$

where

n = an exponent having the value 0.398 for horizontal ducts, and 0.570 for vertical ducts

a = a constant which is some function of the specific gravity of the solids conveyed.

The following equation has been found to fit the data closely for a:

$$a = \frac{s}{s+1}k \tag{4}$$

where

s = the specific gravity of the solids

k = a constant

The complete equations for the two types of transport are:

$$v = 6000\frac{s}{s+1}d^{0.398} \tag{3a}$$

for horizontal ducts and

$$v = 13300\frac{s}{s+1}d^{0.570} \tag{3b}$$

for a vertical system. The velocity v is, of course, in feet per minute and the diameter d is in inches.

The larger margins between the velocity-diameter curves for horizontal transport are due to the tendency of the particles to settle and slide along the bottom surface of the duct. As has already been noted, the critical velocities in a horizontal system are not as sharply defined as those for a vertical system. Equation 3a does not seem to apply as well as Equation 3b to all particles, regardless of size. In some test experiments conducted with fly-ash (mostly iron silicates varying from 1 to 10 microns in size) it was found that there

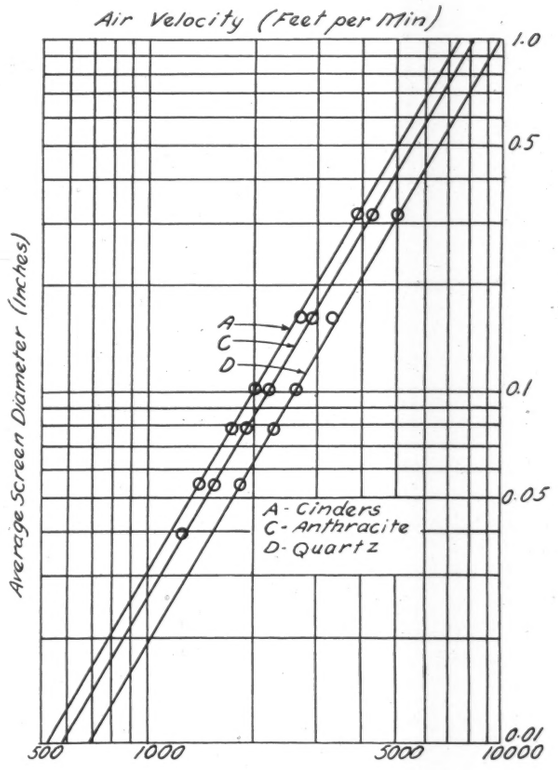


Fig. 3—Data on vertical ducts. (Minimum velocities).

was a tendency for the fines to cling to the bottom of a horizontal duct and to gradually build up. It required a velocity considerably in excess of that which can be deduced from Equation 3a to keep the duct clean. For such substances further work must be done. It is possible that other factors besides density and size are of considerable importance.

The equation for the minimum velocity of transport in a vertical duct would seem to indicate that the velocity varies at a rate greater than the square root of the diameter. This feature may be attributed to the coarseness of the material used and to its dis-symmetry which causes the particles to exhibit spin while in translation.

Inside Surface Temperatures of Windows

Neglecting absorption of radiant heat, the inside surface temperature of window glass under equilibrium conditions is a function of the inside and outside air temperatures and the velocity of air passing over the inside and outside surfaces.

The temperature of the inside glass surface, of course, influences the precipitation of moisture on this surface, since condensation will take place if the surface is below the dew-point temperature.

For practical purposes it is sufficient to regard the velocity of air passing over the inside surface as that due to natural convection, which is usually termed *still air*. The outside wind velocity may be taken as 15 mph, in which case the overall or air to air coefficients, according to the A. S. H. V. E. Guide 1932, are as follows:

Single glass	1.13
Double glass	0.45
Triple glass	0.28

The inside surface coefficient for still air, according to The Guide 1932, is 1.50 Btu per hour per square foot per degree Fahrenheit difference in temperature. The inside surface temperature may readily be computed from these coefficients for any given temperature head. For example, assume the inside temperature to be 70 F and the outside temperature to be —10 F. The inside surface temperature under equilibrium conditions, neglecting absorption of radiant heat, would be obtained as follows:

$$t_s = 70 - \left[70 - (-10)\right] \times \frac{0.45}{1.50} = 46 \text{ F (double glass)}$$

The inside surface temperature may be computed for any other type of glass or set of conditions in a similar manner. The accompanying chart (Fig. *A*) for single, double and triple glass is based on these coefficients which, as stated, are for so-called still air conditions inside and an outside wind velocity of approximately 15 mph.

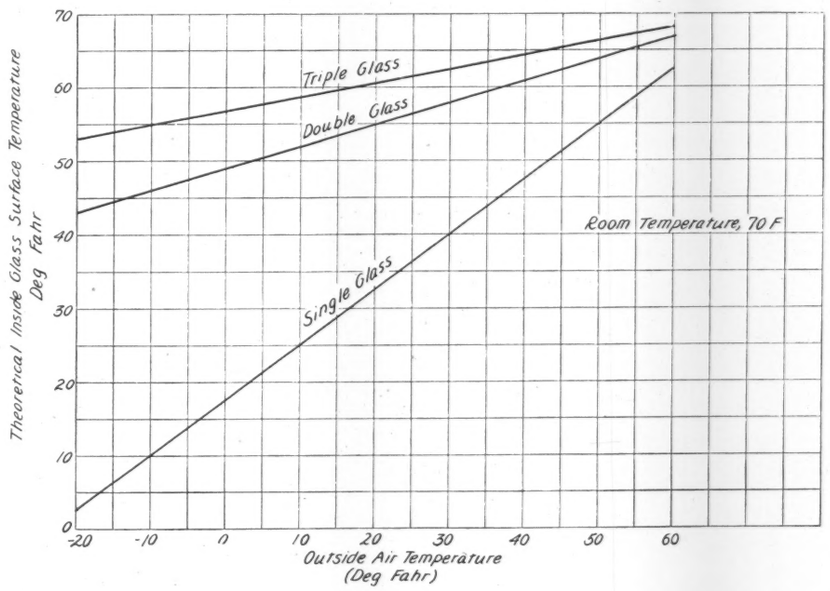


Fig. A — Theoretical inside glass surface temperatures.

Heating by Forced Hot Water

By H. H. Angus*, Toronto, Ontario
MEMBER

FORCED hot water heating systems may include all hot water systems in which the water is circulated by forces other than gravity. Among the methods generally used to produce this accelerated flow are:

1. A reciprocating or centrifugal pump producing a reduced pressure on the return and an increased pressure on the flow line.
2. An injector supplied with steam under pressure which forces the water to flow and at the same time supplies heat to it.
3. Steam blown directly into the rising portion of the flow pipe thus substituting a mixture of steam and water for a solid column of water.

This paper deals primarily with the first method which is the one most generally used. Data regarding pipe sizes, etc., used in the design of forced hot water systems are given in the A. S. H. V. E. GUIDE 1932 and other books. On the other hand, there are many practical points to be taken into consideration and for which very little data are available.

Advantages of Forced Circulation

The main advantages of circulating water by mechanical means rather than by gravity are:

1. Lower capital cost and greater flexibility due to the smaller sizes of pipe and radiators.
2. Boilers, piping, radiators and pumps may be placed at any level to suit conditions on the site, and mains can also be arranged to rise or fall as required by the building conditions. This is of considerable advantage in many cases, for example, where it is desired to place radiators on the basement floor and at the same time not have pipes run under the floor.

The power required to circulate water through a medium-sized forced hot water installation is very small. For example, a system with 5,000 sq ft of heating surface and a 20-ft head would require about ½ hp and with an efficiency of 33 per cent the pump would consume about 1½ hp. Where electric power is reliable and reasonably priced, the cost of operation for the heating season is small. The saving in pipe, fittings, covering, etc., by using a pump will more than cover its cost.

Pumps and Motive Power

On small systems of from 1,000 to 3,000 sq ft of equivalent heating surface, a pump will often prove itself useful where the building is low and spread over considerable area, where garages or similar buildings located at a distance from the main buildings are to be heated from the main plant, or where automatic oil or gas burners are to be used. In these cases it is customary to use a pump to handle large quantities of water at heads of from 12 in. to 30 in. These pumps are connected into the return line and should be designed so that the water can flow through them or through a by-pass when the pump is not in operation. In such systems the pump is either connected to operate simultaneously with the fuel-burning device, or is operated by a hand switch or thermostat at odd intervals during mild weather and more frequently during cold weather. Duplicate units are not warranted, as the probability of breakdown on a simple apparatus of this type is very remote and in any case, there will be considerable circulation by gravity. In general such pumps are driven by fractional horsepower motors and care must be taken in tightening up the glands or the motor will be overloaded.

On larger systems where steam is not available, duplicate electric motor-driven centrifugal pumps are used and if there is any likelihood of interruption of electric service a gasoline engine or other auxiliary is installed. In this case the pump couplings can be built with wide flanges to be used as pulleys for belt drive from the gasoline engine placed between them. Where steam is available it is generally advisable to use steam-turbine-driven centrifugal pumps and to use the exhaust from these in the converters which heat the water.

Where pumps are placed directly under or close to rooms where noise is a factor, they should be properly isolated from the foundations and should be carefully lined up to avoid vibration. In some cases rubber connectors are placed between the pump connections and the piping. Pumps for such locations should be of the slow speed type, and it is possible to eliminate most of the noise from the motor. If, however, the piping around the pump is not installed with proper provision for expansion, it will distort the pump casing when the heat is on and this will often cause trouble with a unit which would otherwise be quiet.

The low head pumps for small installations should be of the screw type or other special design as it is difficult to design a standard type of centrifugal pump for low heads when directly connected to a motor. Pumps for larger installations should be of the enclosed-impeller type with bronze impeller and bronze or monel shaft.

The required capacity of a pump is easily calculated once the drop in temperature in the system is decided. For example, if the heat emission in 240 Btu per square foot of equivalent heating surface and the temperature drop is 20 deg, the pump must have a capacity of 12 lb of water per square foot of heating surface. It will be noted that this is the maximum drop and where pumps are operated at constant speeds this drop in temperature will be only 10 deg in mild weather when the radiators are giving off say 120 Btu per square foot per hour. On small systems the extra cost of a pump for a 20-deg drop is very small and some saving can be made in the heating surface by maintaining a higher temperature in the radiators.

Boilers for Hot Water Systems

In small and medium-sized systems the water is generally heated directly in the boilers. In larger systems,

*Consulting Engineer.

the water is sometimes heated in converters and these are usually supplied with exhaust steam from the circulating pumps, boiler feed pumps and any other auxiliary apparatus.

Heating Surface and Water Temperatures

On medium-sized and large systems, the amount of heating surface required for forced hot water heating should be the same as for low pressure steam. Larger systems are usually operated with closed expansion tanks and in this case the safety valve is usually set so that the mean water temperature will be at least 220 F. As the extremely cold weather is of short duration, there is no good reason why the mean water temperature should not be maintained as high as 270 F during this period, and during ordinary winter weather the temperature would then be about 180 F. Using a system with a 20-deg drop between flow and return, a pressure of about 36 lb gage would be required at the expansion tank. With this system the output would be about 430 Btu per square foot and the amount of heating surface required would be about 55 per cent of the amount required for a low pressure steam system. The standard cast-iron radiator is suitable for use at pressures as high as 60 lb. With the gravity convector type of heating unit the limit is much higher and the tendency should be, where such heating units are installed, to use higher water temperatures and smaller heating units at a considerable saving in capital cost. Water temperatures as high as 240 F have been used to a considerable extent on gravity hot water systems with concealed, fin-type heating units.

City water mains are being used to some extent to take the place of the expansion tank on larger systems. It is not unlikely that the smaller systems using fin-type heating units and high pressure boilers will be connected in the same manner. With the fin or convection-type heating units, the heating effect drops rapidly with the lower range of temperatures when used with low temperature systems. With the high temperature system, the temperature of the water in spring and fall will usually be sufficient to maintain a high rate of output with these radiators.

Piping Layout

Piping for a forced hot water system may be designed on the basis of either the one- or two-pipe systems to suit the building and other conditions, and in general, it will be found that the two-pipe system is the easier to design and install. The building can usually be divided into a number of branch systems and the piping for each of these designed on the reverse-return principle so that in any one circuit the water has the same distance to travel from the boiler to each radiator.

Generally a main supply header is placed in a central location. From this header branch supply and return mains are run to the different sections into which the system is divided. A shut-off valve, also a drip valve and a thermometer, are placed in each line. The flow of water in the different branches is usually adjusted by valves so that the temperatures are the same on all lines. In general it is not necessary to figure the pipe sizes for forced hot water as accurately as for gravity systems as the friction varies as the square of the velocity. For this reason a slight error in figuring pipe sizes usually will not cause any trouble with the uniform heating of the system. Adjusting valves are generally installed but seldom used except where a central system is supplying a number of buildings whose circuits have been designed on the basis of different friction heads.

The friction head for which a given hot water system is to be designed may be calculated if the cost of power is known. It is a question of finding the point at which an increase in the head will result in a cost of power for driving the pump greater than the saving in interest and depreciation charges of the saving in capital cost of pipe. In general it will be found that a velocity of over 5 fps (feet per second) will result in noise in the pipe. This would not be serious for a manufacturing plant but for office or residential work it might be objectionable. For a uniform friction head per linear foot of pipe it will be found that the velocities in the risers, branches to radiators, etc., will probably not exceed 3 fps in the average building, where cost of power is a factor.

Many systems are in use where the radiator branches are of ⅜-in. pipe. Where pipe of this size is used there should be no burrs on the ends of the pipe when cut or the area of the pipe will be greatly reduced. Such piping needs to be installed by skilled men. If ½-in. pipe is the smallest size used, little or no difficulty is likely to be experienced. On systems where both supply and return mains are run in the basement and the highest radiators are several floors above the mains, it is advisable to figure the risers for a low friction. By doing this, the temperatures of the top radiators will be about the same as those on the lower floors. Orifices may be used to balance the friction in the radiators on the different floors but the writer's experience has been that where risers are figured for a low head there will be no necessity for using orifices. Where it is possible to run the supply main in the attic space or above the radiators and the return in the basement the risers may be calculated in the same manner as the remainder of the system.

Connections of risers to mains and radiators are the same as for the steam systems except for provision for expansion and removal of the air from the system. Connections are generally made at the bottom of opposite ends of radiators except for those below the mains which have top supply. Little difficulty will be experienced in securing uniform circulation in all radiators without the use of adjusting valves providing the system has been designed correctly. It is however, advisable to install shut-off valves and drips with hose connections at the base of each riser. On small systems one radiator valve on the supply or return of the radiator is used, but on large systems in well finished buildings a lock shield valve on the other end is advisable so that the supply of water may be shut off in case of a leaky or damaged radiator. In the writer's experience these valves are seldom used.

Freezing of Water in Radiators

One of the objections to the use of hot water for heating is the danger of freezing the water in the radiators. However, in localities where the temperature often drops to —30 F there are practically no troubles from frozen radiators. A small hole is sometimes drilled through the radiator valve so that there will be some cir-

culation even with the valve closed. In some cases, it has been found necessary, where continuous cold weather has stopped building operations, to turn heat into a new building, and this can be done without danger. On one installation with which the writer is familiar, the system was first tried out with an outside temperature of —20 F. As a precaution, only a section was opened at a time so as to secure a rapid circulation, but no trouble was experienced in getting the system into operation, and the building warmed up satisfactorily.

Repairs

The repairs on a hot water system should be less than on a low pressure steam system as the repairs on the hot water system are generally confined to the pumps and boilers or converters. On steam systems there are traps, reducing valves, air vents, vacuum pumps, etc., all of which require cleaning at frequent intervals and some repairs.

The problem of corrosion in steam heating systems is of considerable importance. Vol. XXII of the *National District Heating Association* handbook contains a report on this subject and shows that this factor must be taken into account. Corrosion affects different parts of the heating system but is chiefly evident on the return piping. It is apparently due partly to the condition of the steam entering the system and partly to the leakage of air into the system.

On any system of steam piping there is likely to be leakage of air into the returns as the pressure in this piping will be below atmospheric at least part of the time. Hot water heating offers a solution to this problem as the pressure is always above atmospheric and the water in the system remains for a long time and soon becomes inactive. The small amount of water required for make-up should not affect the system, but in any case this water could be treated before being used.

Comparative Costs

There is a general impression that a forced hot water system costs more to install than a low pressure vacuum system of the better type. As a check on this, the writer made estimates on a heating system for an 8-story hospital building containing about 31,000 sq ft of heating surface. A forced hot water system was installed in the building and has been in successful operation for the last two winters. The system was designed for a friction head of 30 ft and the friction loss per 10-ft run in the circuits averaged about 0.4 ft. A steam system was designed for this building using the same amount of heating surface and an 8-oz total pressure drop, the pipe sizes having been selected from tables given in the A. S. H. V. E. Guide. The total cost of pipe and fittings for the steam system was $5,340 and for the hot water system, $4,197. These prices do not include hangers nor do they include covering which would depend to some extent on whether the risers were exposed or concealed. The traps and valves are not included in the figure for the steam system. On each radiator in the steam system there was a radiator valve and a trap as compared with two valves for each radiator for the hot water system. The steam system also required drips on the ends of mains and at the bottom of risers. There were 70 risers and 630 radiators and the extra cost of valves and traps for the steam system was estimated to be $1,500, thus making a total extra cost of $2,740 for the steam system. These prices are based on starting from the steam and return headers located in a basement room near the center of the building. These figures would, of course, vary in different localities.

The power required to circulate the water is 4 hp theoretical or about 8 hp actual, and the corresponding vacuum pump would require a 3 hp motor. This comparison, which agrees with an independent estimate by the contractor who did the work, covers only the piping, and to this must be added the cost of pumps, heaters, boilers, etc. If the system is not connected with a separate power plant there would be required two electrical motor-driven circulating pumps with possibly an auxiliary gasoline engine. The steam system would require a duplex vacuum pump or two single pumps. The boiler would be the same in each case. The cost of the duplex vacuum pump would be about $1,000 more than the hot water circulating pumps, making a total extra cost of $3,700. If the system were supplied from a power house or central steam plant the steam system would require reducing valves as well as pumps and the hot water system, converters and pumps. In this case the cost of the steam pumps with reducing valves and control equipment for operating the system economically, would be somewhat lower than the cost of the converters and circulating pumps but not sufficiently so to make up for the difference in the cost of piping.

The figures on this installation show that the cost of the forced hot water system is lower than the low-pressure steam system. Figures made on another hospital 6 stories high and having 15,400 sq ft of heating surface in 360 radiators using two converters and two steam-turbine-driven circulating pumps, showed the hot water system to be $500 cheaper than a low-pressure steam installation. It is evident, however, that there would be considerable variation in costs for a building of a different size and shape, but the hot water system would, in many cases, be cheaper. With a steam system floor trenches are sometimes required and these can be eliminated with forced hot water.

Adaptation to Fluctuating Conditions

Forced hot water systems are used extensively in the colder climates where it is necessary to have heat continuously over considerable periods. The use of small piping and the fin-type radiator reduce materially the amount of water in a hot water system and brings it much closer to the steam system in its ability to change rapidly to suit changing conditions. The heat stored in walls, floors, furniture, etc., is considerable and the advantages of rapid increase or decrease in temperature seem to be greatly exaggerated in many cases.

Economizers and Draft

An economizer used with a steam plant offers a resistance to the flow of gases, and lowers the average chimney gas temperature thereby decreasing the available intensity of the draft. Both of these factors result in a relative increase in the required height of the chimney, and in the case of a large plant, may add as much as 20 to 30 ft to the height.

A. S. H. V. E. Standard Code for Testing and Rating Steam Unit Ventilators

Edition of February 1932

COMMITTEE—JOHN HOWATT, *Chairman*, C. P. BRIDGES, S. E. DIBBLE, WARREN EWALD, L. D. HARNETT, G. E. OTIS, R. C. OTT.

A. OBJECT OF THE CODE

1. **The Object** of this code is to provide a method of testing and rating the heat and air output of steam unit ventilators.

B. DEFINITIONS

2. **A Steam Unit Ventilator** for the purposes of this code shall be construed to mean a combined heating and ventilating unit arranged with connections for taking air direct from outdoors and with the heating unit heated with low-pressure steam and with the fan or fans and motor as an integral part of the unit.

3. **The Entering Temperature** for the purposes of this code shall be construed to mean the average temperature in degrees Fahrenheit of the air entering the unit measured at the air inlet.

4. **The Final Temperature** for the purposes of this code shall be construed to mean the average temperature in degrees Fahrenheit of the air discharged from the unit measured at the air outlet.

5. **The Power Input** for the purposes of this code shall be construed to mean the power input to the motor at the stated fan speed and at the stated conditions of rating.

6. **The Steam Pressure** for the purposes of this code shall be construed to mean the gage pressure in pounds per square inch above the standard or normal barometric pressure of 14.7 lb per square inch (29.921 in. of mercury).

7. **Standard Air** for the purposes of this code shall be construed to mean air weighing 0.07495 lb per cubic foot[1] and having a specific heat of 0.2415.

8. **The Total Equivalent Direct Radiation (EDR)** for any condition of operation when supplied with steam at 2 lb gage pressure shall, for the purposes of this code, be construed to mean the heat in the discharged air minus the heat in the entering air, in Btu at a specified inlet air temperature divided by 240.

9. **Surplus or Heating Equivalent Direct Radiation** for the purposes of this code shall be construed to mean difference between the total EDR at a specified inlet temperature and the EDR required to heat the air from that temperature to 70 F.

C. BASIS OF RATING

10. **Rating Factors to Be Specified.** The rating of the unit ventilator shall specify the following:

a. Final temperature at different entering air temperatures.

b. Total EDR at different entering air temperatures.

c. Air delivered by the unit in cubic feet per minute at the *standard basis of rating* with the fans operated at rated speed, with all air being blown through the heating unit and with the standard louvre and grille on the outlet.

11. **The Standard Basis of Rating** shall be as follows:

a. Dry saturated steam at a temperature at the unit corresponding to an absolute pressure of 16.7 lb per square inch (218.5 F).

b. Entering air temperature of zero degrees Fahrenheit.

c. Volume delivered in cubic feet per minute converted to standard air at 70 F.

12. **Rating Tables** for unit ventilators shall contain the following data in addition to the standard rating, for entering air temperatures from —30 F to +60 F:

a. Inlet temperature, degrees Fahrenheit

b. Final temperature, degrees Fahrenheit

c. Total EDR at the specified entering temperature

d. Surplus or heating EDR at the specified entering temperature.

13. **Tolerance.** As there are errors of measurement and inequalities of manufacture, a variation of 2½ per cent in test results shall not be considered excessive.

D. OUTLINE OF TESTS

14. **Heat Output—Air Volume Tests.** This code prescribes tests to determine the heat output in Btu from the weight of condensation, and the volume of air in cubic feet per minute from condensation and the temperature rise of the air passing through the unit, and further prescribes a means for correcting the heat output and temperature rise as obtained under test conditions to any other condition of entering air temperature and steam pressure.

15. **Air Quantity Check Test.** A check of the air quantity determined by means of the heat content of the condensation and the temperature rise of the air passing through the unit shall be made by direct measurement with a calibrated nozzle. The results of the two methods shall agree within 5 per cent before the tests shall be considered correct. However, the results obtained by means of the condensation-temperature rise method shall govern for the purpose of rating under this code.

16. **Receiving Chamber.** The unit being tested shall discharge into a receiving chamber (Fig. 1) in which the air shall be thoroughly mixed in order that the true average temperature of the air may be determined. The conditions within this chamber shall be controlled in such a way that the unit will deliver the same capacity as when under normal conditions of free delivery.

17. **Free Delivery Conditions.** In order to make certain that the tests on the receiving chamber are a faithful reproduction of free delivery conditions the following tests shall be conducted:

a. The condensation of the unit shall be measured dur-

[1] This weight corresponds to dry air at 70 F or air with a relative humidity of 50 per cent at a dry-bulb temperature of 68 F, when the barometric pressure is 29.921 in. of mercury.

Report of Committee presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January 1932. Adopted as a Standard of the Society June 1932.

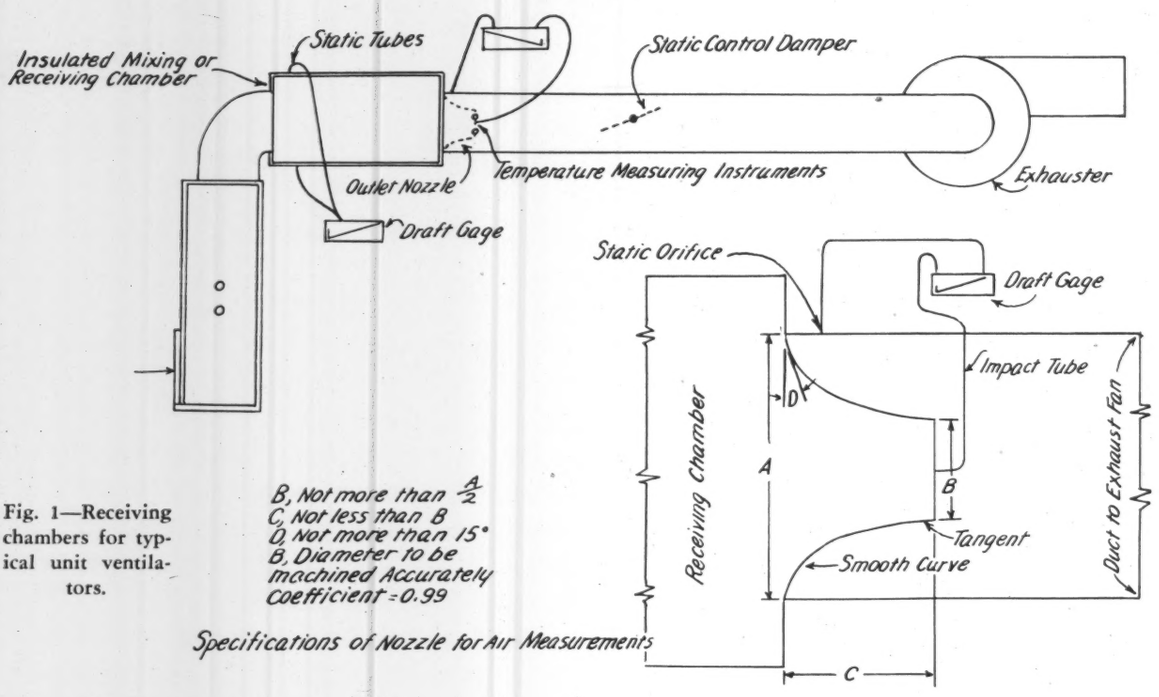


Fig. 1—Receiving chambers for typical unit ventilators.

ing a test under actual free delivery conditions before the unit is connected with the receiving chamber.

b. Three tests shall be made with the unit connected to the receiving chamber, one at a constant pressure of — 0.05 in., one at a constant pressure of zero and one at a constant pressure of + 0.05 in. water column on the chamber pressure draft gage.

During these tests it shall be necessary to measure only the condensation, entering air temperature and chamber pressure. These shall each be of but ½ hour duration. These four tests shall be made at the same fan speed and at entering air temperatures within 4 deg of each other.

18. Correction for Entering Temperature. The condensation measured on these three chamber tests shall be corrected to the entering temperature of the free delivery test in the proportion of the temperature difference between the steam and entering air.

19. Condensation—Chamber Pressure Curve. A curve shall then be plotted from the three tests on the chamber showing condensation against chamber pressure. This curve shall be assumed to be a straight line within this narrow range of chamber pressures. From this curve shall be read the chamber pressure corresponding to the condensation measured on the free delivery test. The draft gage, which records the receiving chamber pressure, shall be held constant at the reading thus determined, during the tests necessary for rating data as described hereinafter. This series of tests for the calibration of the receiving chamber to produce free delivery conditions shall be made for each unit at rated speed. Should the free delivery condensation in any series of tests meet the chamber pressure curve for that series at a point above + 0.05 or below — 0.05 it is an indication that too great an error has been made either in the free delivery test or the check tests on the chamber, which shall be corrected or the test repeated.

E. EQUIPMENT FOR TESTING

20. A Chamber for receiving and mixing the air discharged from the unit shall be provided. This chamber shall be constructed of any suitable material and shall be air-tight and insulated with 2 in. cork, or equivalent.

21. Size of Receiving Chamber. This receiving chamber shall be of such size that the unit to be tested will produce from 20 to 90 air changes per minute.

22. Exhaust Fan. This receiving chamber shall be connected by a duct to an independent exhaust fan of such capacity that it will overcome the resistance of the chamber and connections and produce a zero static pressure at the point where the heater outlet is joined to the chamber.

23. Static Orifices. Two or more static orifices shall be located in the receiving chamber and not in the direct air blast from the unit. These openings shall be connected by air tight tubes to a common draft gage which can be read to 0.005 in.

24. Control of Exhaust. Means shall be provided with which to vary the capacity of the exhaust fan so as to maintain a zero static constant on the draft gage.

25. A Calibrated Nozzle shall be fitted into one wall of the chamber, discharging into the duct leading to the exhaust fan. The outlet opening of this nozzle shall be of such area that the air velocity is not less than 3000 fpm.

26. Instruments shall be located at the point of 3000 fpm minimum velocity for measuring the final temperature, which shall be the average of temperatures taken simultaneously at at least two points in the plane of the nozzle outlet for each square foot of outlet area, but in no case less than four points.

27. A Draft Gage shall be provided for measuring pressures at the nozzle. One side of the draft gage shall be connected to a static orifice located flush with the inner wall of the exhaust duct near the chamber wall. The other side shall be connected to an impact tube arranged to measure the velocity pressure of the air in the nozzle outlet.

28. The Air Handled by the unit shall be disposed of in such a way as to prevent fluctuation in the temperature of the air entering the unit.

29. Steam Shall Be Supplied from a source of sufficient capacity to prevent sudden changes in pressure. The pressure in the supply line up to the throttling valve shall be maintained at approximately 10 lb.

30. The Piping Diagram (Fig. 2) shows the connections and equipment prescribed for supplying steam to the coil and measuring the condensate. All of the fittings and in-

struments shown shall be used and shall be installed in the relative positions indicated.

31. Separators, Steam Throttle Valves, Manometer. Separators shall be of liberal capacity. Steam throttle valves shall be of a type suitable for close control. In selecting the type of manometer consideration shall be given to the fact that condensation will collect above the mercury on the steam pressure side and lend to the error unless compensated for.

32. The Petcock used for air relief shall be not more than ⅛ in. in size.

33. The Water Leg shall have a sight gage so that the water level can be brought to the same point at the time of each reading of the condensate.

34. The Fittings and Piping from the radiator outlet to the thermometer shall be insulated.

35. The Scale for weighing condensate shall be of the beam type capable of being read to 0.25 lb.

36. The Tanks receiving the condensate shall be covered to reduce the loss by evaporation.

37. Temperature Measuring Instruments shall be placed around the intake of the unit in such locations and in such numbers as will reflect a true average of the entering air temperature. All temperature measuring instruments shall be capable of being read to 0.5 deg F or closer and shall be calibrated. When exposed to radiant heat they shall be shielded therefrom.

38. The Heater Casing need not be insulated for the purpose of this code.

39. A Stop Watch shall be used for the accurate timing of readings.

40. A Barometer shall be provided to determine the atmospheric pressure during test.

F. TEST PROCEDURE

41. Preliminary. The steam shall be turned on; all valves and air vents shall be opened wide for a sufficient length of time to blow out all water and air; the fans shall be started; the control shall be set to give the predetermined static pressure in the receiving chamber; the air relief cock shall be set only so that a thread of steam escapes continuously; the draft gage zero shall be checked by disconnecting the static tube; the exhaust damper shall be reset if necessary; the unit shall be warmed until conditions have become stabilized before starting test.

42. Steam Pressure. During the test the steam at the heating unit inlet shall be held constant at 2 lb gage with a superheat of not less than 2 deg.

43. Throttle Valve. An operator shall be in constant control of the throttle valve using the manometer as the reference instrument. Before the test is begun a reference line shall be marked on the manometer corresponding to the 2 lb test pressure corrected for any water in the manometer. During the test the steam valve shall be manipulated in order to hold the mercury steady at this line. In order to introduce the required superheat there shall not be less than 5 lb drop in pressure through the throttle valve.

44. Weighing Tank Tare. The condensate shall be diverted to the waste tank and the tare on the weighing tank determined. The scales shall be free in operation and both scales and tank shall be free from external contacts. The point at which the water level is to be held at each point of reading the condensate shall be marked on the sight gage of the water leg.

45. Starting Test. The steam pressure and temperature shall be checked; the time shall be noted to the second, and the water diverted to the weighing tank—this begins the test.

46. Duration of Test. The test should be continued for one hour during which time all conditions shall be held as nearly constant as possible.

47. The Following Readings shall be taken and recorded at intervals of 10 minutes or less:

a. Steam pressure
b. Steam temperature (at coil inlet and outlet)
c. Air inlet temperatures
d. Air outlet temperatures
e. Weight of condensate for the period
f. Static pressure at unit discharge
g. Watts input to motors
h. Differential pressure at the nozzle

48. Weighing of Condensate. Exactly at the end of each period the condensate shall be diverted to an alternate weighing tank or change bucket. Accumulation of condensate for the period shall be weighed and recorded.

49. Consistent Data. If the data recorded for successive periods are inconsistent or vary beyond a reasonable margin, the test shall be continued until one hour of consistent data are recorded. Care shall be exercised to bring the water level in the sight gage to the original position at each reading and to divert the condensate at the end of each reading period to the next period.

G. COMPUTATION OF RESULTS

50. Chamber Calibration Tests. The condensation obtained on each of the receiving chamber check tests (C) shall be converted by the following formula to its approximate equivalent (C') at the entering air temperature (t'_e) of the test made at the same fan speed under free delivery conditions:

$$C' = C\,\frac{t'_s - t'_e}{t_s - t_e} \qquad (1)$$

where

t_s = the steam temperature of the receiving chamber test
t_e = the entering air temperature of the receiving chamber test
t'_s = the steam temperature of the free delivery test

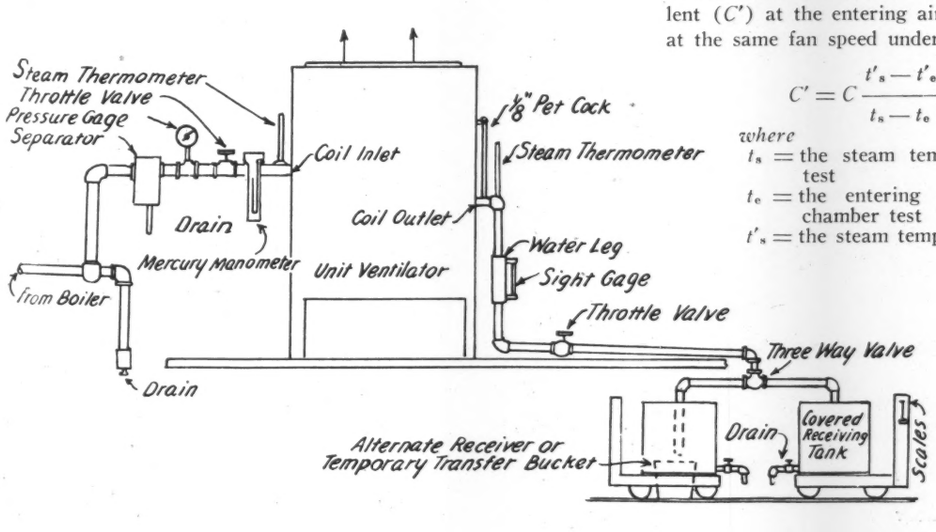


Fig. 2—Piping diagram for unit ventilator test equipment.

51. Check Test of Air Volume. The volume of air, in cubic feet per minute, passing through the nozzle at the final temperature as of test conditions, shall be calculated by the following formula:

$$\text{Volume} = 1096\ AK\sqrt{\frac{VP}{W}} \qquad (2)$$

where

A = the area of the nozzle in square feet
VP = the differential pressure across the nozzle
W = the weight of air at the temperature in the nozzle and the barometric pressure of the test
K = the coefficient of the nozzle

52. Rating Tests. The following result directly from the test data:

t_e = average temperature of entering air, degrees Fahrenheit
t_t = average temperature of leaving air, degrees Fahrenheit
t_r = temperature rise of air, degrees Fahrenheit
t_s = saturated temperature of steam, degrees Fahrenheit
p = pressure of steam, pounds per square inch, gage
b = barometric pressure, pounds per square inch
P = total or absolute pressure of steam = $p + b$
C = pounds of condensation per hour
h_{fg} = latent heat of steam

53. Rating Under Stated Conditions. It is generally impracticable to test units under exact predetermined entering air temperatures and steam pressures. Since, however, it is necessary to rate them under stated conditions, the data obtained from the test may be used for the determination of performance under such desired conditions of rating. For this purpose, the assumptions in Paragraphs 54 and 55 shall be made.

54. Constant Air Volume Assumption. It shall be assumed that the volume of air handled by the fan or fans at the temperature in the fans is constant for a given unit and fan speed regardless of temperature and barometric pressure changes.

55. Temperature Rise Assumption. It shall be assumed that the formula for determining the average temperature rise of the air under free delivery conditions, namely:

$$t'_r = \frac{t_r\,(t'_s - t'_e)}{(t_s - t_e)} \qquad (3)$$

is true for a constant volume of air as well as for constant weight within the range and limits of this code.

56. Formulae for Computing Results. The following formulae may be used to compute results:

$$Q = \frac{h_{fg} \times C}{14.5 \times t_r \times W_e} \qquad (4)$$

$$Q' = Q\,\frac{(460 + 70)}{(460 + t'_e)} \qquad (5)$$

$$t'_r = t_r\,\frac{(t'_s - t'_e)}{(t_s - t_e)} \qquad (6)$$

$$H = 14.5\ Q \times t'_r \times W''_e \qquad (7)$$

where

Q = volume of air delivered, cubic feet per minute measured at inlet temperature and barometric pressure of the test.
Q' = rated volume of standard air delivered, cubic feet per minute
C = pounds of condensation per hour
h_{fg} = latent heat of steam
W_e = weight of air per cubic foot at t_e and at the barometric pressure of the test
W''_e = weight of air per cubic foot at t'_e and at standard barometric pressure (29.921 in. of mercury)
14.5 = 60 x 0.2415 (specific heat)
H = heat output, Btu per hour

57. Example. Assume the following data as having been obtained from a test: $C = 90$ lb; $t_e = 30$ F; $t_t = 111$ F; $t_r = t_t - t_e = 81$ F; barometric pressure of the test = 29.00 in. of mercury; $t_s = 219$ F; $h_{fg} = 966$ Btu per pound; rpm = 490. Substituting the proper values in Formulae 4, 5, 6 and 7:

$$Q = \frac{966 \times 90}{14.5 \times 81 \times 0.07858} = 942 \text{ cfm}$$

$$Q' = 942\,\frac{460 + 70}{460 + 0} = 1085 \text{ cfm (for } t'_e = 0)$$

$$t'_r = \frac{81\ (219 - 0)}{219 - 30} = 94 \text{ F (for } t'_e = 0)$$

$$H = 14.5 \times 942 \times 94 \times 0.08636 = 110{,}900 \text{ Btu per hour.}$$

58. Sample Table. A complete performance table may be calculated for different entering temperatures in accordance with Paragraph 57, as follows:

t_e	t_r	Btu	Total EDR	Surplus EDR
-10	98	118,300	493	91
0	94	110,900	462	118
10	90	103,900	433	114
20	85	96,200	401	165
30	81	89,700	374	190
40	77	83,600	348	213
50	72	76,600	319	230
60	68	71,000	296	253

Pressure Measurement

Atmospheric pressure is usually measured by a mercurial barometer which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The atmosphere, pressing on the exposed top of the mercury in the cistern, supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage,* and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage.* The most common type of these gages contains a flexible hollow brass tube of oval cross section known as a *Bourdon tube.* When subjected to unequal inside and outside pressures, this tube tends to straighten out, and a pointer motivated by this straightening indicates on a suitably graduated scale the pressure difference.

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage.* It is essentially a U tube containing either water, kerosene, alcohol or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pressure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

Cincinnati Chapter Appoints Annual Meeting Committees

The newly organized Cincinnati Chapter of the Society is planning to entertain the members at the 39th Annual Meeting of the Society, January 23 to 25, 1933, and the Officers and Board of Governors announce the personnel of the Committee on Arrangements:

W. C. Green, *General Chairman*
H. N. Kitchell, *Vice-Chairman*
J. J. LaSalvia, *Chairman,* Reception Committee
E. B. Royer, *Chairman,* Registration Committee
H. M. Rudio, *Chairman,* Entertainment Committee
H. E. Sproull, *Chairman,* Banquet Committee
R. B. Breneman, *Chairman,* Ladies Committee
I. B. Helburn, *Chairman,* Transportation Committee
J. J. Braun, *Chairman,* Publicity Committee
C. J. Kiefer, *Chairman,* Finance Committee

Each of the Sub-Committee Chairmen will select his assistants.

The recently elected officers of the Chapter are: K. A. Wright, *President;* H. M. Kitchell, *Vice-President;* G. B. Houliston, *Secretary-Treasurer;* C. J. Kiefer and E. B. Royer, *Board of Governors.*

They extend a cordial invitation to all members of the Society to come to Cincinnati in January.

A. I. E. E. Appoints 1932 Committees

Announcement has been made by the *American Institute of Electrical Engineers,* 33 West 39th St., New York, N. Y., of Committee Appointments for the year commencing August 1, 1932. In the group are 25 general committees and 18 technical committees.

Westinghouse Enters Air Conditioning Field

A complete organization for engineering development and sales of air conditioning apparatus has been announced by Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa. According to the announcement of J. S. Tritle, vice-president and general manager, the following appointments have been made:

J. W. Speer, formerly Manager Machinery Electrification, has been appointed Manager Commercial Air Conditioning Products, Diversified Products Department, in charge of the sale of commercial and railway air conditioning apparatus; R. C. Cosgrove, Manager, Refrigeration Division, will head the domestic sales; W. C. Goodwin, Supply Engineering Department, is in charge of engineering and development.

Welding Encyclopedia

The eighth edition of the *Welding Encyclopedia,* published by the Welding Engineer Publishing Co., Chicago, is now available.

The significant advances which have been made in the applications of welding to production have been accompanied by interesting developments in the various processes. In general, these developments have been in the direction of increased production efficiency and improved weld quality. The results attained have encouraged a more serious interest in precision methods of control, through more systematic use of inspection and test procedures. These are subjects which have been treated with special care in the preparation of this new edition.

The contents include the following: Welding Encyclopedia, Electric Arc Welding, Oxy-Acetylene Welding, Electric Resistance Welding, Thermit Welding, Cutting Metals, Boiler Welding, Pipe Welding, Structural Steel Welding, Tank Welding, Training Operators, Rules and Regulations, Heat Treatment of Steel, Charts and Tables, Catalog Section.

N. D. H. A. Handbook

The second edition of the Handbook of the *National District Heating Association* has just been issued in flexible leather binding with 473 pages of technical data and an equipment section of 54 pages, with indices to make a total of 538 pages. The Educational Committee, in issuing this volume, states that the purpose is to provide an authoritative and complete manual of practice for the use of those actively engaged in district heating work. It is pointed out by the Committee that it has made use of the many papers and reports in the association's *Proceedings* and has taken data from the files of many operating companies and from various other sources.

The technical section has been divided into 12 chapters, which present an historical outline of district heating service, general data on calculating heat transmission, selecting heating elements, heat transfer from pipes, and comfort data.

In Chapter 3, building heating methods are described and in Chapter 4 equipment and accessories are covered. The remaining chapters treat the subjects of steam distribution, district heating plants, meters, hot water heating, corrosion, rates, sales activities and the economical use of steam.

Many of the figures on heat transfer through building materials, infiltration, and the comfort data have been taken from The A. S. H. V. E. Guide.

The personnel of the committee responsible for this publication consists of: J. H. Walker, *Chairman,* D. S. Boyden, *Vice-Chairman,* Wm. J. Baldwin, Jr., A. R. Mumford, Earle Shultz, G. D. Winans, John W. Meyer and the Handbook Editor was C. H. B. Hotchkiss.

Death of Charles J. Steim, Jr.

Friends of Charles J. Steim, Jr., Buffalo, N. Y., were shocked to learn of his death on July 25, 1932. At the time of his demise he was Branch Manager of the Buffalo office of the C. A. Dunham Co.

Mr. Steim attended the Grade Schools in Pittsburgh, Pa., the College Preparatory at Shadyside Academy, and studied two years at Carnegie Institute of Technology, Pittsburgh. After leaving school, he occupied various positions with the Pennsylvania State Highway Department, E. P. Bock, James H. Corbett, W. F. Devendorf and Co., Rochester, N. Y., Standard Manufacturing Co., Toledo, Ohio, and Samuel Sloan and Co. of Rochester.

He was well liked by his many friends, and his untimely death closed a career of outstanding success. Mr. Steim became a member of the Society in 1923 and the Officers and Council join in expressing their sorrow to his family and associates.

VENTILATION STANDARDS ADOPTED

A meeting of the Board of Tellers to count the ballots of members voting on the Report of the Committee on Ventilation Standards was held at the headquarters office of the Society in New York on August 12. The report of the tellers submitted by W. L. Fleisher, *Chairman,* C. H. B. Hotchkiss and M. E. Durkee showed a total of 546 ballots received of which 501 were legal votes—383 in the affirmative and 117 in the negative and 1 incomplete.

Death of George P. Smith

Friends of George P. Smith were shocked to learn of his sudden death on Monday, August 8. He was well known in Chicago having served for many years as branch manager of the Herman Nelson Corp., Moline, Ill. Mr. Smith was born November 7, 1874, at Joliet, Ill.

His experience in the heating profession began in 1906 and for three years he was in charge of estimating and construction for Poehur and Dielman. From 1909 to 1912 he was manager of the heating department for the Western Plumbing Supply Co. and held a similar position for the next six years with the Mine and Smelter Supply Co. In 1918 he became identified with the Herman Nelson Corp., and for several years served as sales representative in Salt Lake City. During the past 10 years he has been in Chicago where he was an active member of the Illinois Chapter and made a host of friends.

Mr. Smith joined the Society in 1922 and his death is greatly regretted by the Officers and Council who extend their sincere sympathy to the members of his family who survive, his widow Alice M. Smith, his mother Mrs. C. J. Smith and his brother and sister Thomas V. and Mrs. J. W. Fitzgibbons. Funeral services were held for Mr. Smith August 11 and he was buried in the city of his birth, Joliet, Ill.

Death of Thomas H. Monaghan

It is with deep regret that the death of Thomas H. Monaghan on Thursday, August 11, is recorded. As president of Robert Gordon, Inc., through his membership in several societies and associations and because of his personal qualities, Mr. Monaghan's friendships were numerous. Funeral services were held August 14 at Iron Mountain, Mich.

Born in Philadelphia, May 30, 1883, Mr. Monaghan graduated from the Central High School in Kansas City, Mo., in 1900. From 1900 to 1911 he was associated with various firms in Kansas City as draftsman, estimator or engineer. In 1911, as engineer for Robert Gordon, he moved to Chicago.

He became president of Robert Gordon, Inc., in 1915 and was closely connected with the construction of a number of outstanding buildings, his firm being the heating and ventilating contractors for the Merchandise Mart, the Union Station, the Field building and the Parcel-Post building—among many others —in Chicago.

Mr. Monaghan married Ethel Higgins of Iron Mountain, Mich., in 1916. He became a member of the Society in 1914, and the Officers and Council join his many friends in extending their sincere sympathy to his widow and his two sons, Thomas Higgins and Gordon Hampton, who survive.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 2 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by September 15, 1932, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
SIGMUND, RALPH W., Dist. Mgr., B. F. Sturtevant Co., 913 Provident Bank, Cincinnati, Ohio	G. B. Houliston C. J. Kiefer	E. B. Royer
WECHSBERG, OTTO, Pres. and Gen. Mgr., Coppus Engineering Corp., Worcester, Mass.	E. Elliot W. T. Jones	H. C. Robinson M. J. Rodman

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

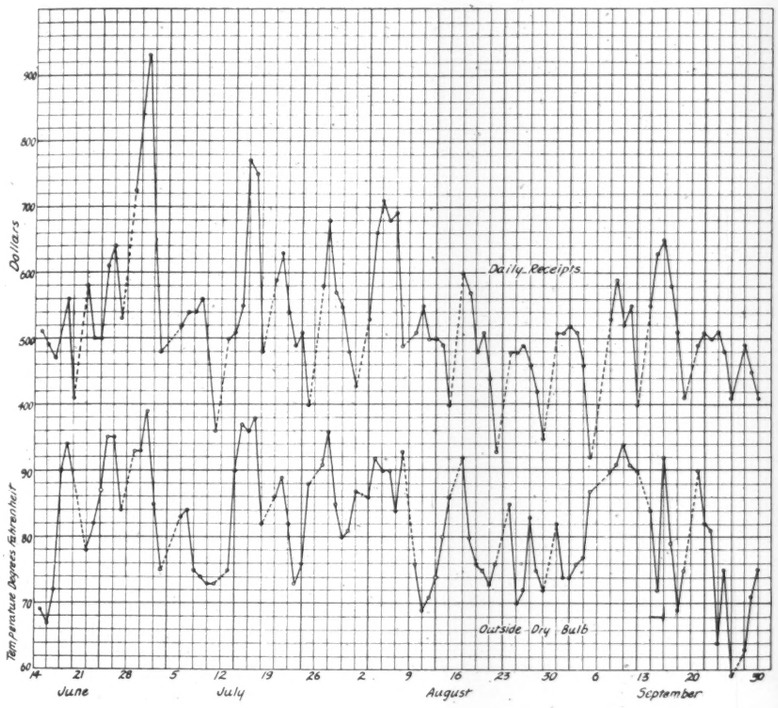
MEMBERS

BERRY, FEROL W., Engr., Hall-Neal Furnace Co., Indianapolis, Ind.

OAKEY, WILLIAM E., Cons. Engr., Babcock Mfg. Co., Leonardsville, N. Y.

RUDIO, HARRY M., Sales Engr., Carrier Engineering Corp., 1501 Carew Tower, Cincinnati, Ohio (*Reinstatement*)

. . . . Does Cooling Pay?



. . . . . It does in the restaurant whose daily receipts for the summer of 1931 are plotted here! Note how with high outside temperatures the public wants comfort and seeks an air-conditioned restaurant. Outside temperatures were taken from official records and the income figures from the auditor's books.

THE above chart, which shows the actual daily receipts of an air-conditioned Chicago restaurant for the summer months of 1931 and the outside dry-bulb temperatures, indicates emphatically that cooling does pay. Note how closely the upper curve of daily receipts follows the lower curve of outdoor temperatures. On warm days—when the public is seeking comfort—many patrons who would normally go to a nearer, but uncooled, eating place come from considerable distances to dine in comfort.

For example, the chart shows that on June 17 the outside temperature was 94 F and $560.00 was taken in by the restaurant. When the temperature dropped to 88 F on June 21, the day's receipts were only $410.00. On July 1, with an outside temperature of 99 F, receipts were $930.00 and on July 3, with a temperature of 75 F, they were but $480.00.

The dotted lines indicate that the restaurant is closed on Sunday and invariably before them a shrinkage is shown due to Saturday noon and afternoon.

Conventions and Expositions

Safety Congress: October 3-7, Washington, D. C. National Safety Council, 20 N. Wacker Drive, Chicago, Ill.

American Gas Association: Annual convention, October 10-14, Atlantic City, N. J. American Gas Association, 420 Lexington Ave., New York City.

American Society of Mechanical Engineers: Annual meeting, Nov. 30-Dec. 4, Engineering Societies Bldg., New York City. Secretary, Calvin W. Rice, 29 W. 39th St., New York City.

National Power Show: Dec. 5-10, Grand Central Palace, New York City. Manager, Charles F. Roth, International Exposition Co., Grand Central Palace, New York City.

Recent Trade Literature

Air-Conditioning Units: Motor Wheel Corporation, Lansing, Mich.; 12-page manual of ratings, specifications, and data for oil-burning forced-air heating, humidifying and air circulating units for small buildings.

Air Filters: Owens-Illinois Glass Company, Toledo, Ohio; 16-page booklet describing and illustrating filters made of glass wool, and presenting diagrams showing arrangement of assembled units, installation details, etc.

Air Washers: Buffalo Forge Company, Buffalo, N. Y.; four-page bulletin of ratings and a description of unit air washers.

Alloys: The American Brass Company, Waterbury, Conn.; eight-page pamphlet discussing an alloy of copper with silicon and manganese as a material for tanks and pressure vessels; tanks from 1 to 20,000-gal. capacity and several water heaters are illustrated.

Alloys: The International Nickel Company, Inc., 67 Wall St., New York City; 12-page technical bulletin presenting information on the corrosion resistance of metals against sulphuric acid, particularly nickel alloys. Numerous curves and tables are included.

Boilers: Kewanee Boiler Corporation, Kewanee, Ill.; four-page bulletin giving features, dimensions and